Iron and Steel Engineer

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J. F. KELLY, *Editor*

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Coming Meetings and Papers

PITTSBURGH DISTRICT SECTION

FEBRUARY 18, 1931

"VICES AND APPLIANCES FOR STEEL MILL ROLL NECK LUBRICATION," by Lubrication System Manufacturers. Each paper to be based on an actual Steel Mill installation. Meeting held under the auspices of the Lubrication Division of the A. I. & S. E. E. Meeting will be held in the Blue Room, William Penn Hotel. Technical Session starts at 2:00 P. M.

MARCH 13, 1931

JOINT MEETING A. I. & S. E. E. AND A. I. E. E.

9:30 A. M.

"CONVERSION AND DISTRIBUTION OF GENERAL PURPOSE D.C. POWER IN LARGE INDUSTRIAL PLANTS," by R. D. Abbiss, Assistant Chief Engineer, Carnegie Steel Company, Braddock, Pa., and D. C. West, Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa.

"INTERCONNECTION BETWEEN THE DUQUESNE LIGHT COMPANY AND THE DAVISON COKE & IRON COMPANY," by G. E. Dignan, Chief Engineer, Davison Coke & Iron Co., Pittsburgh, Pa., and R. L. Kirk, Assistant to System Development Manager, Duquesne Light Company, Pittsburgh, Pa.

"INTERCONNECTION FOR POWER BETWEEN PUBLIC UTILITIES AND LARGE INDUSTRIAL USERS," by F. O. Schnure, Superintendent of Electrical Department, Bethlehem Steel Company, Sparrows Point, Md.

"ABSORPTION OF BY-PRODUCT POWER," by A. Hoefle, Engineer, Toledo Edison Company, Toledo, Ohio, and W. T. Woodmancy, Superintendent of Power, Interlake Iron Corporation, Toledo, Ohio.

2:00 P. M.

"THE USE OF ELECTRICITY IN LARGE ANNEALING FURNACES," by J. C. Woodson, Manager of Industrial Heating Engineering Dept., Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

PHILADELPHIA DISTRICT SECTION

FEBRUARY 7, 1931

"ANNEALING IN ARTIFICIAL ATMOSPHERES IN ELECTRIC FURNACES," by Horace Drever, Gas Equipment Engineering Corp., Philadelphia, Pa.

MARCH 7, 1931

'SOME IMPRESSIONS OF A WELDING SUPERVISOR IN A STEEL MILL—IN A MANUFACTURING PLANT," by J. C. Cumberland, Supt. of Welding, A. S. Cameron Steam Pump Works, Phillipsburg, N. J.

BIRMINGHAM DISTRICT SECTION

FEBRUARY 28, 1931

"HOW AND WHY VACUUM TUBES ENTER THE ELECTRICAL FIELD," by W. C. White, Vacuum Tube Engineering Department, General Electric Co., Schenectady, N. Y.

MARCH 28, 1931

"LATEST DEVELOPMENTS IN UNIVERSAL PLATE MILLS," by E. S. Lammers, General Engineer, Westinghouse Electric & Mfg. Co., Atlanta, Ga.

CLEVELAND DISTRICT SECTION

FEBRUARY 27, 1931

"VARIOUS ASPECTS OF THE FUEL BURNING PROBLEM IN MODERN STEAM PLANTS," by H. G. Reichard, District Manager, Combustion Engineering Corporation, Cleveland, Ohio.

CHICAGO DISTRICT SECTION

FEBRUARY 10, 1931

"ROLLING MILL LUBRICATION," by E. S. Glauch, Mechanical Engineer, Joseph Dixon Crucible Company, Jersey City, N. J.

EDITORIAL

A. I. & S. E. E. Convention
and
Iron and Steel Exposition
JUNE 15, 16, 17, 18 and 19, 1931

WHY SHOULD I ATTEND THE A. I. & S. E. E. CONVENTION?

FIRST—To meet the Engineers and Executives of other plants and to discuss informally my routine problems and to exchange opinions and ideas in connection with the design, installation, operation and maintenance of equipment used to process and produce Iron and Steel.

SECOND—To attend the Technical Sessions at which time will be discussed the developments during the last year, such as—

The Twin Motor Drive. Cold Strip Mills. Lubrication of Steel Mill Equipment. Anti-Friction Bearings in the Iron and Steel Industry. Improvements desired in Combustion Control Apparatus.

Improvements in Production Methods with respect to Wire and Wire Drawing Machinery.

The Practical Side of Metallurgical Heating Applications. Cables for Cranes. Lighting in Steel Plants. Standardized Symbols for Control Apparatus. Welding in the Iron and Steel Industry. Air Conditioning in Steel Plants.

WHO ATTENDS THE A. I. & S. E. E. CONVENTION?

Electrical, Combustion, Mechanical, Lubrication, Safety Engineers, Master Mechanics, Chief Engineers, Operating Superintendents, Presidents, General Managers.

WHAT IS THE IRON AND STEEL EXPOSITION?

The Iron and Steel Exposition is a show of Steel Mill equipment held in connection with the Annual Convention of the A. I. & S. E. E. and is owned, sponsored and controlled by the Association of Iron and Steel Electrical Engineers; the only engineering society in the Industry. The first exhibit took place in 1919 at St. Louis, Mo.

WHY SHOULD AN EQUIPMENT MANUFACTURER BE REPRESENTED IN THE IRON AND STEEL EXPOSITION?

The object of the Iron and Steel Exposition is to acquaint engineers with the status of the art in reference to the various lines of apparatus used in the Industry. Accomplishments by the use of improved devices, new applications, etc., largely form the subject of the papers and discussions of the Annual Convention. The apparatus used in improvement of the quality of product and economy of operation develop the urge to make improvements. Exhibits and Expositions develop this urge into a real desire as concrete suggestions and ideas are presented in the equipment displays. Your equipment will do certain things for the Steel Industry. Show the men who actually spend the gigantic appropriations just what it will do and how it will do it. These are the men who will decide just whose equipment will be used. Are they entirely familiar with your products? Put your equipment in the Iron and Steel Exposition where the engineer can operate it, examine it and thoroughly satisfy himself just how it will meet his needs in the field.

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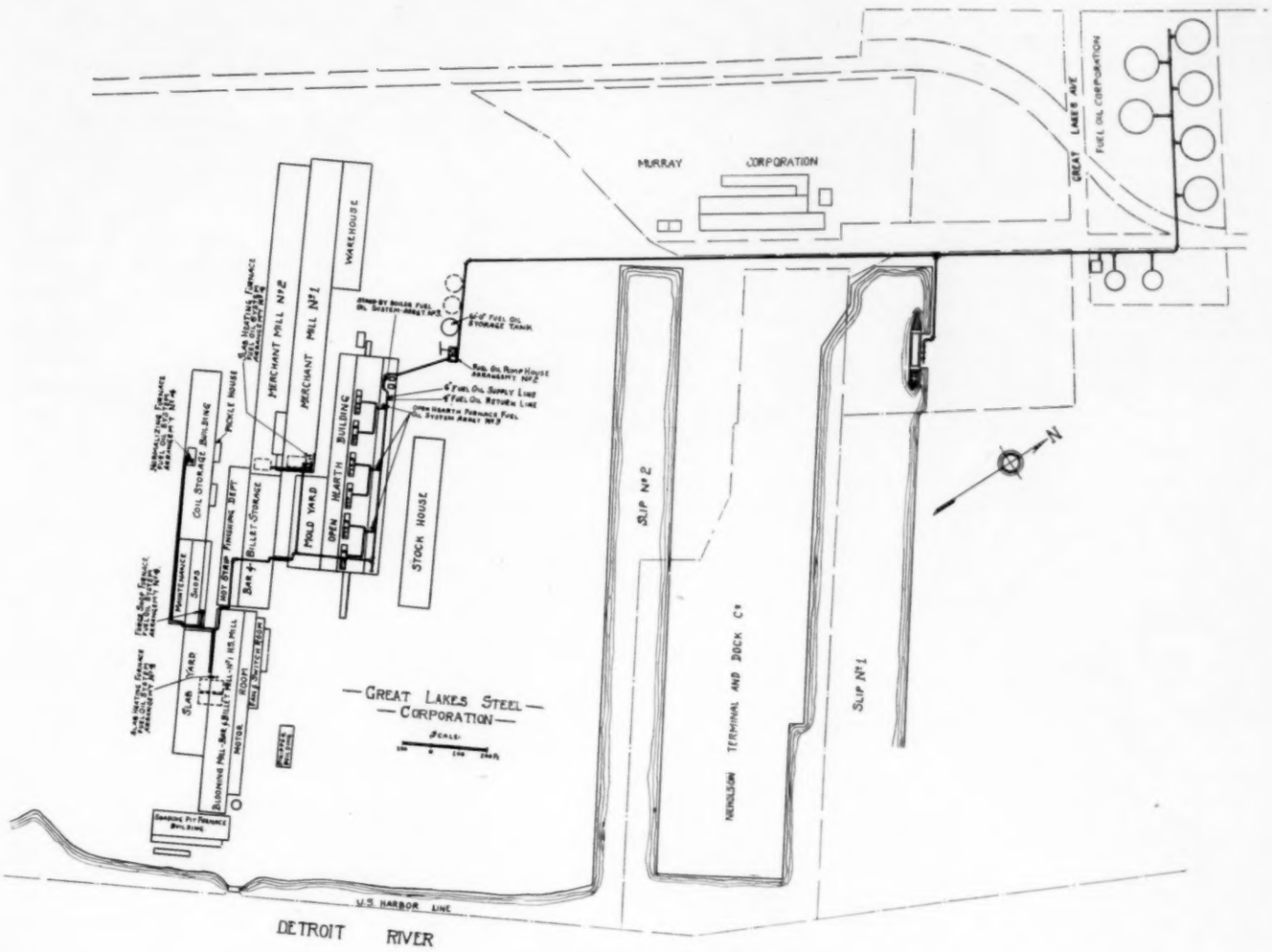


ARRANGEMENT No. 1

Fuel Oil System

Great Lakes Steel Corporation Ecorse, Michigan

By J. A. CLAUSS*

The successful operation of a steel-making plant depends largely on the proper selection and economical consumption of fuels in the Open Hearth and the various reheating furnaces. The fuel to be used is selected, not only for economic reasons, but also because of its availability and the reliability of the source of supply.

In the case of the Great Lakes Steel Corporation's plant, oil was selected as the primary fuel for the Open Hearth, Billet Reheating, and Forging furnaces. The Soaking Pit furnaces are the only heating units, so far installed in the plant, using another fuel. These latter furnaces have been arranged for producer gas operation.

Local conditions and economic necessity prevented the immediate provision for an auxiliary fuel; provision has been made, however, for the installation of gas producers adjacent to all main heating units. Because of this lack of an auxiliary fuel, the design and arrangement of the fuel oil supply system was considered of prime importance in order that no serious interruption in the fuel supply would be encountered and that the large amount of fuel oil would be economically distributed to the various heating units.

The general arrangement of the fuel oil system from the source of supply to the individual furnaces is shown in Arrangement No. 1 which is shown above. Oil is received by boats docking in Slip No. 1 and is pumped either direct to the Great Lakes Steel Corporation's 61'0"x29'0" storage tank or to the Fuel Oil Corporation's storage tank farm. The Fuel Oil Corporation is at present under contract to maintain a definite supply of oil in the storage tank of the Great Lakes Steel Corporation and

* Chief Engineer, Great Lakes Steel Corporation, Ecorse, Michigan.

to maintain a sufficient reserve supply at the tank farm to cover the fuel requirements during the closed navigation season. The oil stored at the tank farm is pumped as required through the connecting pipe line, approximately 2500 feet in length, to the Great Lakes Steel Corporation's storage tank. It may be noted also that, at some future time, oil may be received direct from boats docking in Slip No. 2, which serves the Great Lakes Steel Corporation's property, and pumped to suitable storage tanks located in the plant.

The 61'0"x29'0" storage tank has a capacity of approximately 600,000 gal., which it is estimated, would be sufficient to supply the fuel requirements for about seven (7) days for the furnaces already installed or in the process of construction. This storage capacity will prevent any serious interruption in the fuel supply due to temporary failure of the source of supply. Further insurance against the possibility of interruption is provided by the installation of equipment to handle and store fuel oil received by rail. Space has also been provided, and the oil-handling equipment so arranged, that additional storage tanks may be installed at such time as the growth of the plant demands such additional storage capacity.

Arrangement No. 2 shows diagrammatically the relation of the storage tank to the pumping equipment and the arrangement of the main fuel oil pumping station. Oil is taken from the 600,000 gal. storage tank through a 12" line to the suction header of the pressure pumps for distribution throughout the plant. The pumps, the battery of oil heaters, the meters, valves, and piping constitute the primary portion of this fuel oil system. A secondary element of the system is made up of a piping arrangement for unloading oil received by rail, the 25,000 gal. tank for receiving this oil from the tank cars, and the electric-driven rotary pump for pumping the contents of the receiving tank into the 600,000 gal. storage tank.

The selection of the proper kind and size of fuel-handling equipment for a plant in the process of construction is, in itself, a difficult task. Assumptions must be made as to the nature of the fuel oils available and equipment must be selected of proper size to take care of future expansion and also operate economically at the capacities required by the initial installation.

A study of the market disclosed a widely varying range of viscosities in the available fuel oils. This condition necessitated the installation of equipment capable of handling these oils in order to take advantage of the market. The following analysis would appear to be an average for the heavier fuel oils now available:

Viscosity	236 Sec. @ 210°F. Saybolt Universal 1035 Sec. @ 100°F. Furol
Flash	280°-300°F.
Fire	340°-360°F.
Sulphur	Under ¾ of 1%
Cold Test Pour	Plus 25° to 50°F.
Precipitation with Naptha & 90% Benzol	0 to ½ of 1%
B.T.U.	154,000 per gal.
Gravity Be	13.6

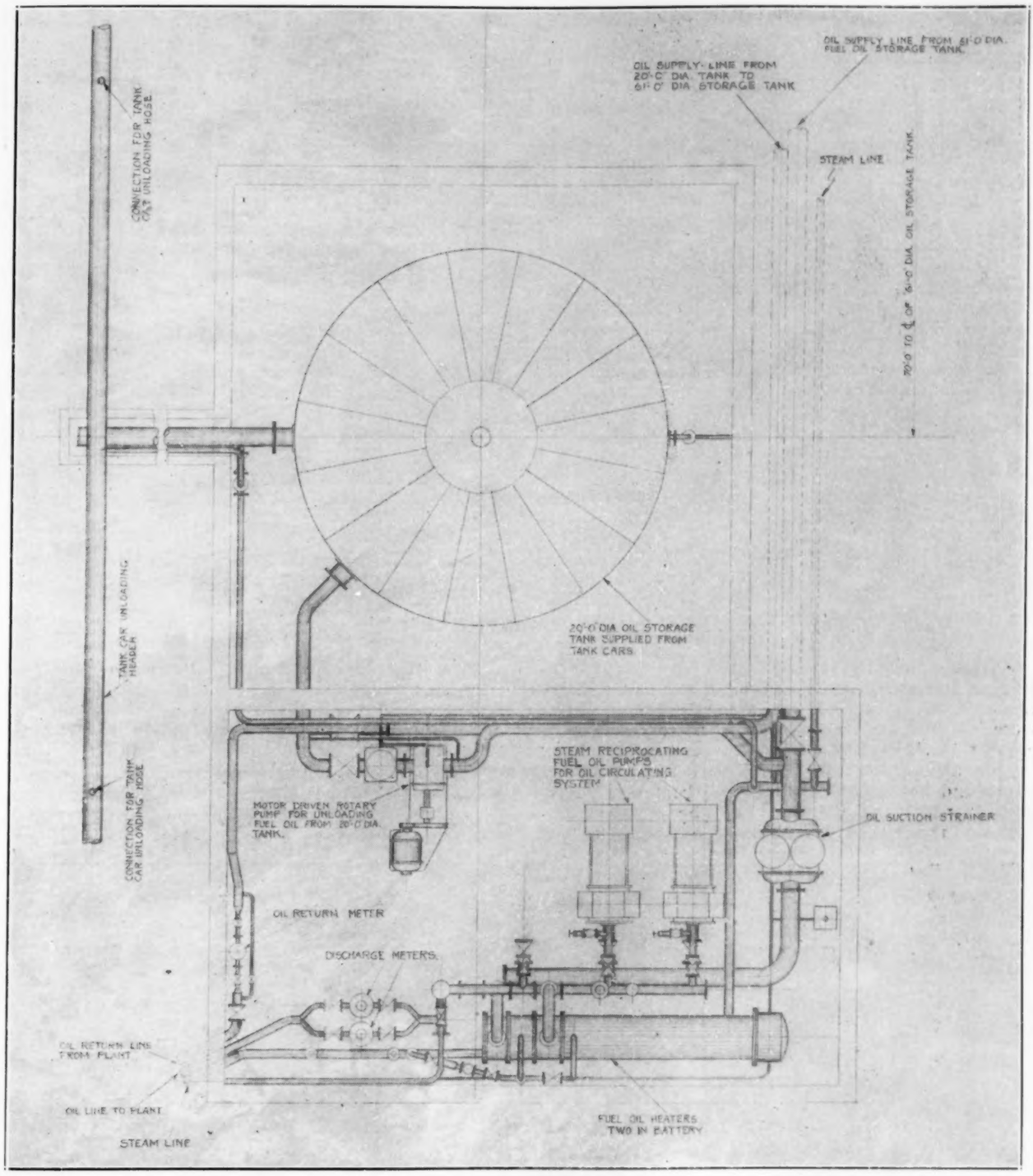
A main recirculating supply system was installed to take care of these heavier and viscous oils. This piping system is shown in Arrangement No. 1 and consists of a 6" supply main for the outgoing oil from the pumping station and a 4" line for the returning oil. These pipes leave the main pumphouse through a pipe tunnel, approximately 200'0" in length, connect the pumphouse with the Open Hearth building, and extend the length of this building which, for the present installation, is 770'0" in length. Provision has been made for the extension of this piping system as more furnaces are added, and outlets were provided to take care of all possible contingencies.

Pipe sizes, pumping and heating capacities were based upon 33% recirculation of the oil. The purpose of this recirculation is to keep the supply lines open to prevent possible deposition of sediment and to aid in maintaining a uniform temperature of the oil in the system.

An extremely flexible oil conditioning system was installed in the main pumping station as shown in Arrangement No. 2. Two (2) pumps and heaters were selected for the initial installation; one (1) pump and heater have sufficient capacity to take care of the present requirements, and the second pump and heater are for standby use. By the installation of a third pump and heater, two (2) pumps and heaters will take care of the ultimate requirements of the plant, leaving one (1) pump and heater as standby units. The pumphouse piping, pumps, and heaters are so arranged that any one, or all of the units, may be placed in service as needed.

Since the viscosity of oils may be raised by temperature, and the capacity of a piping system depends upon the viscosity of the oil and the initial pumping pressure, the heaters, automatic temperature controls, pumps and pressure controls were selected within such ranges as deemed necessary to economically handle the oils available. The heaters were installed with temperature controls and have a capacity to maintain the temperature of the outgoing oil within the range of 90° to 200°F. The pumps and pressure controls were selected of such capacity as to be able to maintain an initial pressure within the range of 100 lb. to 175 lb. per square inch. Steam-driven reciprocating pumps were installed for this service because they are capable of handling oils of widely varying viscosities and lend themselves admirably to automatic pressure and capacity control. These pumps were installed with automatic pressure controls, overspeed controls, and, for added insurance, a relief valve was installed in the discharge of each pump.

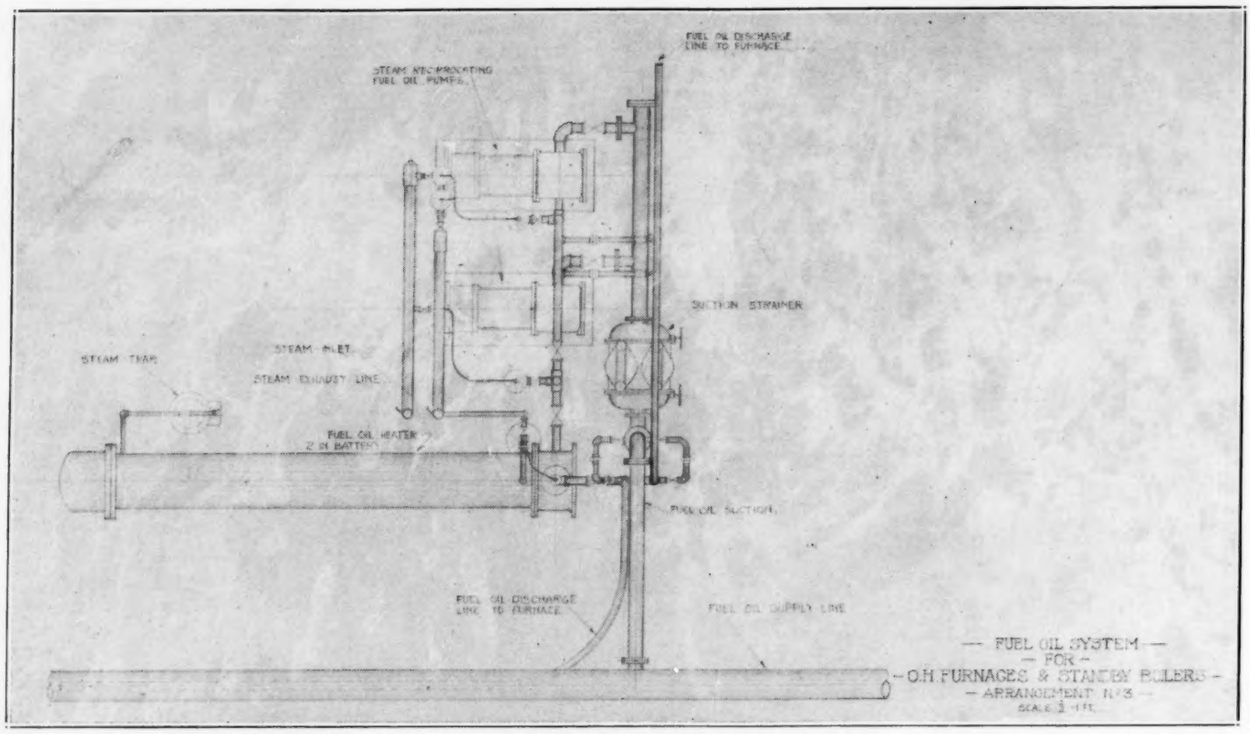
In addition to the above-described provisions for heating the oil, all fuel oil supply lines were enclosed with a parallel steam line within an insulating covering. The steam lines, which are wrapped with the oil lines, aid in reducing the viscosity of the oil in the pipes so that circulation can be easily started, but are not relied upon to heat the oil any appreciable amount after circulation has begun. Steam coils were provided in the main storage tank to prevent the freezing up of the oil in the tank. Provision was also made to return the heated oil from the recirculating system to the main storage tank, by which means a workable tank temperature may be maintained, thereby improving the action of the


OIL SUPPLY LINE FROM 61'-0" DIA. FUEL OIL STORAGE TANK
OIL SUPPLY LINE FROM 20'-0" DIA. TANK TO 61'-0" DIA. STORAGE TANK
STEAM LINE
CONNECTION FOR TANK CAR UNLOADING HOSE
TANK CAR UNLOADING HEADER
CONNECTION FOR TANK CAR UNLOADING HOSE
20'-0" DIA OIL STORAGE TANK SUPPLIED FROM TANK CARS
STEAM RECIPROCATING FUEL OIL PUMPS FOR OIL CIRCULATING SYSTEM
MOTOR DRIVEN ROTARY PUMP FOR UNLOADING FUEL OIL FROM 20'-0" DIA. TANK.
OIL SUCTION STRAINER
OIL RETURN METER
DISCHARGE METERS
OIL RETURN LINE FROM PLANT
OIL LINE TO PLANT
STEAM LINE
FUEL OIL HEATERS TWO IN BATTERY

ARRANGEMENT No. 2

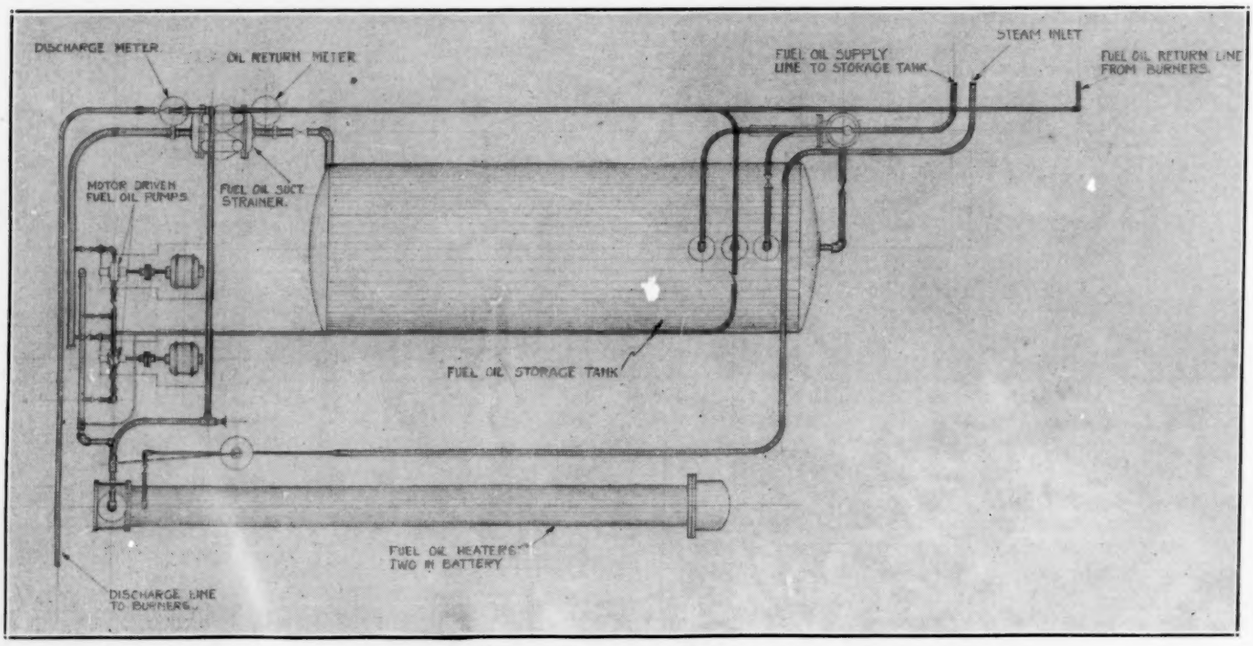
pumps by keeping the oil at the pump suction in a fluid condition.

To secure the proper atomization of the available fuel oils, heating and pumping equipment was installed at each furnace unit throughout the plant. By thus dividing the heating and pumping of the oil into the two stages the most flexible system was provided and it is capable of handling, not only the heavy oils, but also the lighter oils, if they are available, or cracking oil tars. The separation of the system into two stages also permits each furnace unit to be supplied with fuel oil at the most satisfactory pressure and temperature for the economical operation of the particular unit. This method also creates a considerable saving in power and heating costs, as only that oil is heated and pumped to a high temperature and pressure as is needed at the respective furnaces.


FUEL OIL DISCHARGE LINE TO FURNACE
STEAM RECIPROCATING FUEL OIL PUMPS
SUCTION STRAINER
STEAM TRAP
STEAM INLET
STEAM EXHAUST LINE
FUEL OIL HEATER 2 IN BATTERY
FUEL OIL SUCTION
FUEL OIL DISCHARGE LINE TO FURNACE
FUEL OIL SUPPLY LINE
— FUEL OIL SYSTEM —
— FOR —
— O.H. FURNACES & STANDBY BOILERS —
— ARRANGEMENT No 3 —

ARRANGEMENT No. 3

Arrangement No. 3 shows the arrangement of the heating and pumping equipment at the Open Hearth furnaces. A group of two (2) pumps and heaters were installed between each two (2) furnaces of the present six-furnace plant. Provision has been made in the arrangement for the installation of a third pump and heater. One pump and heater serves for each furnace, and the third will be installed as a standby unit. The arrangement of piping is such that either one or both of the heaters and pumps may serve one or both of the adjacent two furnaces. The arrangement of piping also provides for a recirculation of approximately 5% of the oil pumped through the heaters. This recirculation of oil prevents the clogging of pipes about the unit and improves the temperature of the oil by keeping a continuous flow of oil through the heaters when the furnace burners are shut off for any cause. The


DISCHARGE METER
OIL RETURN METER
FUEL OIL SUPPLY LINE TO STORAGE TANK
STEAM INLET
FUEL OIL RETURN LINE FROM BURNERS
MOTOR DRIVEN FUEL OIL PUMPS
FUEL OIL SUCT. STRAINER
FUEL OIL STORAGE TANK
FUEL OIL HEATERS TWO IN BATTERY
DISCHARGE LINE TO BURNERS

ARRANGEMENT No. 4

system outlined in Arrangement No. 3 is similar to that installed for the two (2) 500 B.H.P. Standby Boiler Units. The essential parts of this second-stage oil conditioning unit consists principally of two (2) oil heaters with temperature controls, each having a capacity to heat 500 gal. of oil per hour to temperatures varying from 225° to 300°F., two (2) steam-driven reciprocating pumps with pressure controls of equal capacity and capable of maintaining an initial oil pressure of from 150 lb. to 250 lb. per square inch, an oil strainer, and necessary piping. All of the second-stage oil conditioning units in the Open Hearth plant take their oil directly from the main recirculating system.

The main fuel oil supply lines to the furnace burners are taken from the recirculating systems and terminate at the burner control stands on the Open Hearth Charging Floor. These operating stands are located near the furnace door controls in the X conventional manner and consist of valves for controlling and reversing the burners, an oil meter, and the steam and oil pressure gauges.

The second-stage conditioning systems for the reheating furnaces are shown in Arrangement No. 4. These systems differ from those installed at the Open Hearth furnaces only in that they are supplied with oil from the main supply system through dead-end lines, because the main recirculating system extends only through the Open Hearth building. These dead-end lines are well insulated and paralleled with steam lines which prevent the oil from solidifying in the lines. Provision has also been made to drain this piping system and blow the pipe lines free of oil by the use of steam at such times as the oil is shut off for a considerable period of time.

Arrangement No. 4 shows the oil as entering a suitable receiving tank from the supply line through a float regulating valve which maintains a fixed level of oil in the tank. The receiving tank is vented and the vent is protected by means of a shut-off valve in order to prevent the overflow of oil in the event that the regulating valve fails to close. Each of these systems consist essentially of the vented receiving tank, electric-driven rotary pumps, heaters, meters, and the necessary regulating valves and piping. In each system provision was made for recirculation, and the pumps and heaters were selected to secure the pressures and temperatures required for the proper atomization of the oil in the respective burners. In each installation provision was made for standby heaters and pumps in order to provide insurance against any possible interruption of service due to the failure of this equipment. Also the capacities of the receiving tanks were determined so that any temporary interruption of the main supply system will not interfere with the continuous operation of the various individual units.

Throughout all of the main supply and individual furnace fuel oil system a complete installation of meter equipment was provided. Meters were installed to record the total amount of oil pumped from the main pumphouse, and each individual system was provided with meters to record the amount of oil consumed at the individual units.

STEEL MILL LUBRICATION CONFERENCE

Auspices

LUBRICATION ENGINEERING DIVISION

OF

ASSOCIATION OF IRON AND STEEL ELECTRICAL ENGINEERS

WEDNESDAY, FEBRUARY 18, 1931

William Penn Hotel ***Pittsburgh, Pa.***

PROGRAM

2:00 P. M.

DEVICES AND APPLIANCES

FOR

STEEL MILL ROLL NECK LUBRICATION

The Lubrication System Manufacturers have been invited to present a paper covering Roll Neck Lubrication. Each paper to be based on an actual Steel Mill installation and the Lubrication Committee has suggested that the following points be covered in each Manufacturer's paper:

Type of mill and material rolled on mill.
Location of the System; that is, how far is the system from the stands?
Are pipes carrying lubricant underground, overhead, etc.?
Are pipes insulated?
How long has system been in operation on this particular installation? Is it automatic or manually operated?
How is system operated—by air, water or electricity?
Is the Lubricating System power, independent or is it dependent on the regular plant source of supply?
What arrangements are provided for the quick change of rolls?
Description of the distribution of grease on necks.
Very brief description of control of lubricant at neck.
How is lubricant put into lubricating system from manufacturer's container?
Effect of climatic conditions on lubricating system.
Power savings effected by lubricating system.
Lubricant savings.
Savings in bearings.
Savings in maintenance and upkeep of equipment.
Safety features.
Improvement of product.
Saving in man hours in application of lubricant.
Cleanliness.

This Meeting on Devices Only—Roll Neck Lubricants Will Be Discussed at Second of This Series of Meetings.

Plastic and Elastic Deformation

By E. KIEFT*

Introduction

It is a well known fact that the compression-resistance of wrought iron and mild steel increases to unknown values when we compress plastic material. The rate of increase becomes very great when the height becomes very small as compared with the diameter, this is due to the rapid increase of expansion or horizontal resistance. The writer observed that the resistance per square inch increased from 50,000 to 230,000 lbs.

The applied force works vertically and is transmitted by deformation or displacement of the molecules into horizontal direction. It is therefore necessary that we have to consider not only compression work but also expansion work (the latter is omitted by some investigators) when we try to derive a formula in which the tensile or compression strength of steel is taken as a constant during the whole operation. The assumption that S is constant makes it necessary to determine certain initial conditions as S is dependent on the shape of the material. Different ratios of sectional area to height of test pieces produce different resistance per square inch. It is therefore that the strength of steel is tested on test pieces complying with standard or conventional dimensions.

The friction losses between the block and dies are considered to be included in the expansion work in the following derivations by taking $r^2H=R^2y=$ a constant volume and by the assumption that the initial height is nearly equal the diameter. The formulas may be applied between the range $H=.5$ to 2 diameters. Two blocks of equal volume but different height will show different shapes when reduced to the same final height *(h)* and therefore it is difficult to put all the variations into a simple formula. The difficulty is that S is dependent upon the ratio of diameter and height. On the other hand it is possible to measure maximum pressure *Pmax*, which corresponds to the pressure at the end of the operation as indicated by formula 7, which shows that S becomes the only unknown in the formula after the others are measured and therefore S can be calculated for different ratios of diameter to height of different test pieces and applied for blocks of similar dimensions.

Only certain shapes can be obtained by compressing steel, due to a fixed relation of cohesion between the molecules. This relation is dependent upon temperature, therefore we find different plastic relations and consequently different resistances at different temperatures.

The compression of steel tends to increase its density, while drawing of steel tends to reduce its average density, therefore we cannot consider the rolling process as equivalent to either one of the two processes, but merely as a combination of the two. The density of steel appears to remain constant after the ingot has been reduced to bloom or billet by passing thru four or five roll passes.

The necessity of considering the rolling process a combination of compressing and drawing is confirmed by *fig. 2* and *3* of the paper "Rolling Mill Analysis," which was read by the writer during the A. I. & S. E. E. convention in Buffalo and published in the June, 1930, convention issue of the Iron and Steel Engineer. The Formulas as applied in the application on four different mills indicate that the deformation resistance becomes the sum of $F_0=2P_2 \sin\frac{\theta}{2}$ due to rolling resistance and $F_1=2.3\,A_2S\log_{10}\frac{H}{n}$ due to elongation resistance.

The deformation of a block of steel requires a minimum when the displacement of steel is equal in both directions lateral and longitudinal *fig. 1.*

In case of deformation by means of rolls we displace mostly in longitudinal direction, which means more work than in case of *fig. 1.*

When we put a penny through a pair of small rolls then it changes from circular to elliptical shape *fig. 2*, this confirms the theory in "Rolling Mill Analysis."

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Plastic and Elastic Deformation.

by E. KIEFT

The solution of problems pertaining to plastic and elastic deformation involves the application of the law of least resistance or least work.

1. Plastic Flow of hot steel.

Let W = the amount of work that has to be done when compressing a cylindrical block of steel (Fig. 1) of initial radius r and height H to another cylindrical shape of radius R and height h.

The amount of displacement of metal is a minimum, because deformation from a circular shape to non circular shape requires that some particles have to be moved farther away (Fig. 2).

Experience teaches that all natural changes occur along the paths of least resistance. This law is nothing but a consequence of the law of action equal opposite reaction. All deformation work by compression involves compression work in the direction of the y-y axis and expansion work in the x-z plane, we have then:

$$W = W_c + W_e$$

W_c = compression work.
W_e = expansion work.
$W = -\int P dy$
the minussign indicates a decrease of y when W increases.

Without mathematical proof it is evident that the elements of the shaded area of the ellipse are farther away from the small circle than the elements of the shaded area of the large circle.

This proves that the displacement work becomes a minimum when the shape is maintained. This is only possible with homogeneous metal.

Plastic and Elastic Deformation.

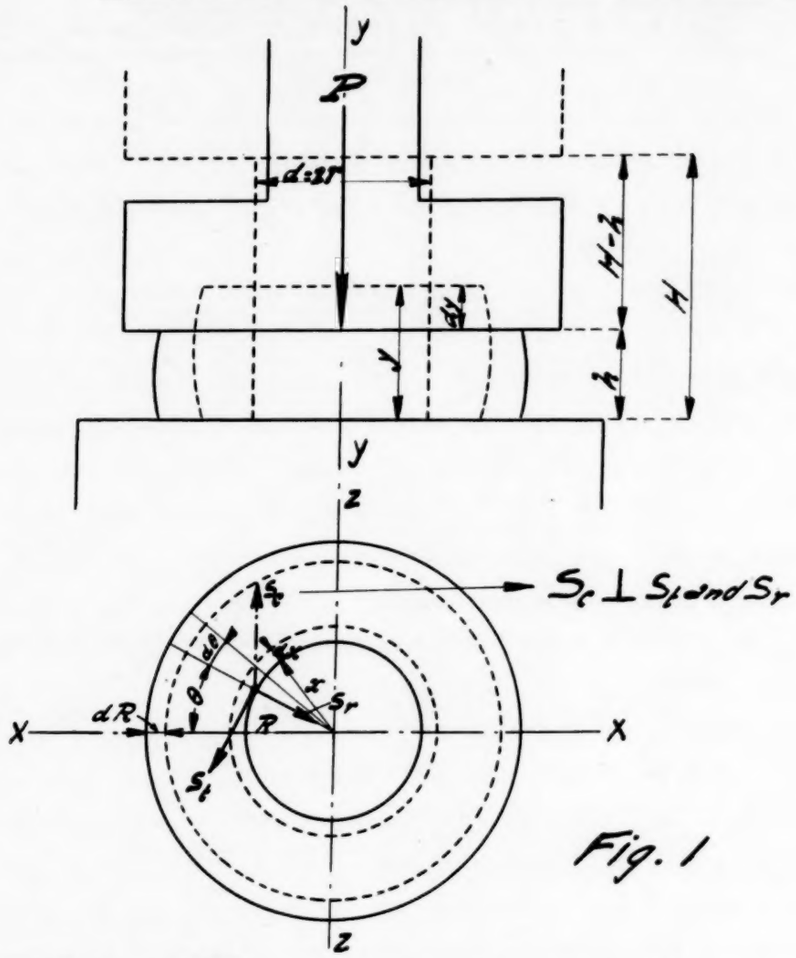


Fig. 1

$\pi r^2 H = \pi R^2 y$ = a constant volume
H equal or less than 2d

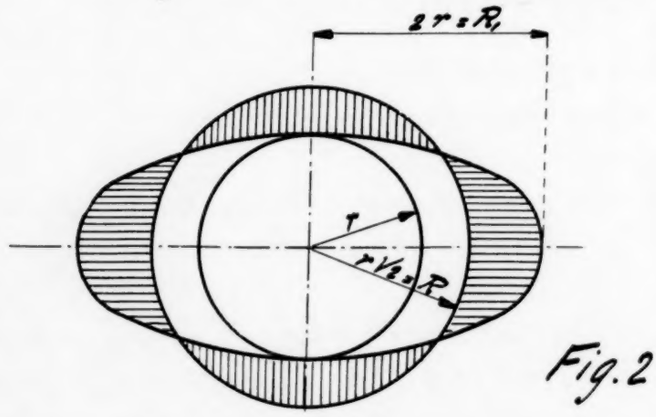


Fig. 2

area of ellipse = $R_1 r \pi = 2 r^2 \pi$
" " large circle = $R^2\pi = 2 r^2 \pi$
" " small circle = $r^2\pi$
Hence area ellipse = area large circle

Plastic and Elastic Deformation.

The expansion involves radial stress S_r and tangential stress S_t. For hot steel $S_c = S_r = S_t$. For cold steel $S_r = S_t = 1.7 S_c = 1.7 S$.

When compressing rolled steel we may assume that the density is practically constant, because the metal has been subjected to high compression during the rolling process.

By differentiation of the fundamental equation for plastic flow we find:

$$dW = dW_c + dW_e = -P\,dy \qquad (1)$$

for hot steel we may substitute:

$$dW_c = -\pi R^2 S\,dy = \frac{-V}{y} S\,dy = VS\frac{-dy}{y}$$

the expansion dR works against the stresses S_t and S_r (Fig 1.)

hence $dW_e = S\iint x\,d\theta\,dx\,\frac{x}{R}\,dR = S\frac{dR}{R}\int_0^{2\pi} d\theta \int_r^R x^2\,dx = \frac{2\pi S(R^3-r^3)}{3}\frac{dR}{R}$

S = ultimate tensile strength of steel in lbs/sq. inch

V = volume $= \pi R^2 y$ in cub. inch.

the product $x\,d\theta\,dx$ represents a small area of the section as shown in fig. 1.

the ratio $\frac{x}{R}$ indicates that the expansion work is proportional with the distance from the axis y-y, that is for $x = r$ we have $dW_e = 0$ and for $x = R$

$$dW_e = S\iint R\,d\theta\,dR\frac{R}{R}\,dR = \frac{2\pi(R^3-r^3)S}{3}\frac{dR}{R}$$

$dx = dR$ when $x = R$, $dx = 0$ when $x = 0$.

$V = \pi R^2 y$ = volume = constant

differentiating both sides of the equation gives

$$d(\pi R^2 y) = \pi R^2 dy + 2\pi y R\,dR = dV = 0$$

or $$2y\,dR = -R\,dy$$

therefore $\frac{dR}{R} = -\frac{dy}{2y}$ substitution this value of $\frac{dR}{R}$ into the expression referring to expansion work gives $dW_e = \frac{2\pi(R^3-r^3)S}{3}\frac{-dy}{2y}$ and by integration between the limits of $y = H$ and $y = h$, we find

$$W_e = \frac{2\pi(R^3-r^3)S}{3}\int_H^h \frac{-dy}{2y} = \frac{\pi(R^3-r^3)S}{3}\int_h^H \frac{dy}{y} = \frac{\pi(R^3-r^3)S}{3}\log_e\frac{H}{h}$$

or $$W_e = VS\left(\frac{R}{3h} - \frac{r}{3H}\right)\log_e\frac{H}{h} \qquad (2)$$

Plastic and Elastic Deformation.

similarly $W_c = VS\int_h^H \frac{dy}{y} = VS \log_e \frac{H}{h}$ (3)

The total work performed by the force P is equal

$W = 2.3\, VS\left(1 + \frac{R}{3h} - \frac{r}{3H}\right)\log_{10}\frac{H}{h}$ (4) $\left(\log\frac{H}{h} = 2\log\frac{R}{r}\right)$

the factor 2.3 converts $\log_e$ to $\log_{10}$

$W = 2.3\, A_2 S\left(h + \frac{R}{3} - \frac{r^2 h}{3H}\right)\log_{10}\frac{H}{h}$ (5)

from relation $-P\,dy = dW_c + dW_e$ we find

$$P = \frac{dW_c + dW_e}{-dy} = \frac{-\pi R^2 S\,dy - \frac{\pi S(R^3 - r^3)\,dy}{3y}}{-dy}$$

therefore $P = \pi R^2 S + \frac{\pi S(R^3 - r^3)}{3y}$

$$P = \pi R^2 S + \pi R^2 S\left(\frac{R - \frac{r^3}{R^2}}{3y}\right)$$

$r^2 H = R^2 y$ or $\frac{r^2}{R^2} = \frac{y}{H}$

hence $P = \pi R^2 S\left(1 + \frac{R}{3y} - \frac{r}{3H}\right)$ (6)

The maximum pressure corresponds to the minimum value of $y = h$

hence $P_{max} = \pi R^2 S\left(1 + \frac{R}{3h} - \frac{r}{3H}\right)$ (7)

The force P as found by means of formula 7 is the pressure exerted by a testing machine at the moment when the machine stops and also of any press when it is at the end of its operation or stroke. This gives us an easy means of checking the formula 7 when S is known.

We find then that a hot cylindrical block of steel requires 7200 tons pressure at the end of the stroke when the stress 3.33 tons per sq. inch, temp. 2000°F, carbon .75.

initial diameter of block 18" or 9" radius
final " " " 32.8" " 16.4" "
initial height " " 10"
final " " " 3"

These results check with actual tests made on a 10000 ton press.

Plastic and Elastic Deformation.

2. Plastic Flow of cold Steel.

The previous formulas hold fairly well for hot steel but tend to give too low results for cold steel.

From actual testdata the writer found that P_{max} follows the relation

$$P_{max} = \pi R^2 C S \left(1 + \frac{R}{3y} - \frac{r}{3H}\right) \qquad (8)$$

or

$$P_{max} = \frac{V}{y} C S \left(1 + \frac{R}{3y} - \frac{r}{3H}\right)$$

$C = 1.7$ for low carbon steel. $S = 50{,}000$ lbs/sq. inch

$\pi R^2 = A_2 = \frac{V}{y}$ = final area in square inch.

The factor C was found when testing steel blocks varying from $\frac{1}{2}$" to $1\frac{1}{2}$" diameter and having the initial height equal to the initial diameter. 1)

The fact that a body is said to be plastic when the form is permanently altered when the stress exceeds the elastic limit makes it evident that the formulas applying to hot steel need modification in case of cold steel, because the ultimate strength determines the value of S in case of hot steel, while S may vary between the elastic limit and ultimate strength in the case pertaining to cold steel.

The following definitions, which are due to Maxwell give the demarcation of viscous flow from plastic flow:

If the form of the body is found to be permanently altered when the stress exceeds a certain value, the body is said to be soft or plastic and the state of the body when the alteration is just going to take place is called the limit of perfect elasticity. If the stress, when it is maintained constant, causes a strain or displacement in the body which increases continually with the time, the substance is said to be viscous. When this continuous

1) Forgoing formula's 7 and 8 include frictional resistance between press and block.

Plastic and Elastic Deformation.

alteration of form is only produced by stresses exceeding a certain value, the substance is called a solid, however soft it may be. When the very smallest stress, if continued long enough, will cause a constantly increasing change in form, the body must be regarded as a viscous fluid, however hard it may be.

Thus a tallow candle is much softer than a stick of sealing wax; but if the candle and the stick of sealing wax are laid horizontally between two supports, the sealing wax will in a few weeks in summer bend with its own weight, while the candle remains straight. The candle is therefore a soft (or plastic) solid, and the sealing wax a very viscous fluid.

What is required to alter the form of a solid is sufficient force, and this, when applied, produces its effect at once. In the case of a viscous fluid it is time which is required, and if enough time is given the very smallest force will produce a sensible effect, such as would be produced by a very large force if suddenly applied.

Thus a block of pitch may be so hard that you cannot make a dint in it with your knuckles; and yet it will, in the course of time, flatten itself out by its own weight and glide down hill like a stream of water.

3. Elastic Flow.

Within the elastic limit we may consider the deformation of the shape of things as elastic flow.

The law of least resistance or better the law of least work is employed when trying to solve problems which are statically indeterminate when the load is applied nonsymmetrically on a table with four legs as shown in fig. 3.

Plastic and Elastic Deformation.

In this case we have also the application of the law of elasticity according which the compressions are proportional the reactions.

Let the linear compressions equal x_1, x_2, x_3 and x_4 corresponding to the reactions R_1, R_2, R_3 and R_4 then $\frac{x_1}{R_1} = \frac{x_2}{R_2} = \frac{x_3}{R_3} = \frac{x_4}{R_4} = C =$ = constant $= \frac{L}{AE}$ in which

L = length of legs in inches.
A = crosssectional area of leg in square inches.
E = modulus of elasticity in lbs per sq. inch
E = 30,000,000 for steel.

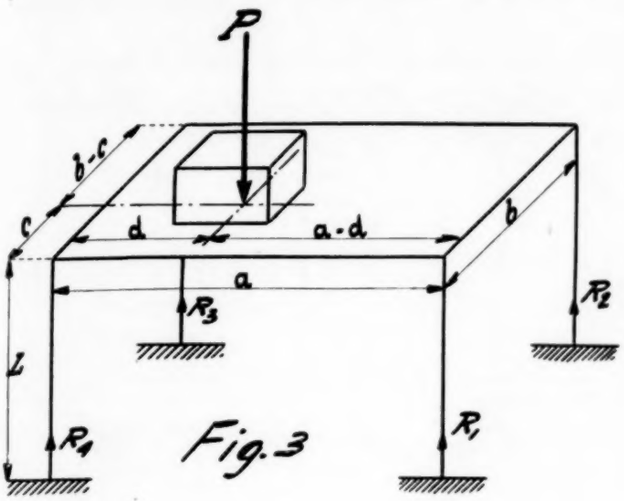


Fig. 3

the compressionwork becomes then

$$W = (R_1 x_1 + R_2 x_2 + R_3 x_3 + R_4 x_4)\frac{L}{2}$$

$$= C(R_1^2 + R_2^2 + R_3^2 + R_4^2)\frac{1}{2} \qquad (9)$$

By method of statical moments we eliminate R_2, R_3 and R_4 and find by putting

$$\frac{d}{a} = m, \quad \frac{b-c}{b} = n, \quad \frac{bd-ac}{ab} = k, \quad m+n+k = y \text{ and } m^2+n^2+k^2 = z$$

$$W = C(4R_1^2 - 2yPR_1 + zP^2)\frac{1}{2} \qquad (10)$$

differentiation of R_1, with respect to W gives

$$\frac{dW}{dR_1} = C(8R_1 - 2yP)\frac{1}{2} \quad \text{and} \quad \frac{d^2W}{dR_1} = 4C$$

which becomes a minimum for $8R_1 - 2yP = 0$

Hence $R_1 = \frac{y}{4}P \frac{(m+n+k)P}{4} = \frac{[ab+2(bd-ac)]P}{4ab}$ (11)

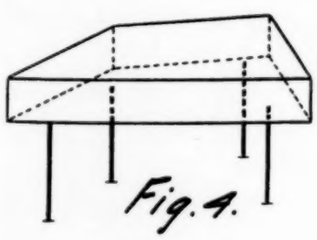
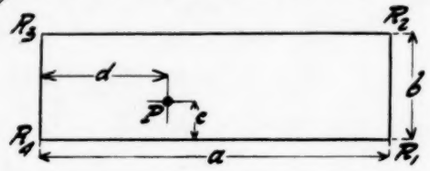


Fig. 4.

The above method can also be applied for finding the reactions produced by a weight as shown in fig. 4.

Plastic and Elastic Deformation.

It is possible that one of the reactions becomes negative or zero as follows from following tabulations for given values of a, b, c, d and P.



Given					Found			
P	a	b	c	d	R_1	$R_2 = mP - R_1$	$R_3 = R_1 - kP$	$R_4 = nP - R_1$
40	5	4	1.2	1.5	10	2	10	18
40	5	4	1	2	13	3	7	17
40	5	4	.5	1.5	13.5	−1.5	6.5	21.5
40	5	4	2.8	1	0	8	20	12

$$m = \frac{d}{a} \; ; \quad n = \frac{b-c}{b} \; ; \quad k = \frac{bd-ac}{ab}$$

The case in which R_2 becomes negative means that the corresponding leg is in tension when connected to the floor, otherwise it indicates that the three other legs carry all the reactions and the problem reduces itself to the simple statical solution of finding the reactions produced by a load on a table supported by three legs.

The case in which the weight of the top of the table is taken into consideration can be solved in similar manner even if we allow for deflection of the table top. In most cases the top is of uniform weight and the total reaction becomes then equal to the reaction due to load P plus one quarter of the weight of the table.

The sum of the reactions must be equal to the load plus the table weight, regardless of deflection and kind of material.

Plastic and Elastic Deformation.

Another application of the theory of elasticity and law of least work we find in Fig. 5, which represents a steel frame fixed at the bottom and subjected to a force 2P.

The frame may be cut along the line A-A and we find that it requires a force Q and a moment M, in order to prevent any distortion of the shape, that is the

deflection $f_A = 0$

the direction of Q is horizontal at A or

$\tan \theta_A$ = zero (fig. 7)

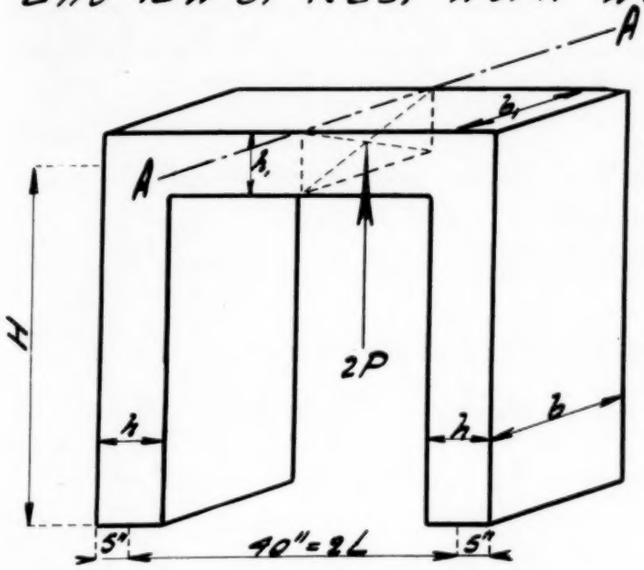


Fig. 5
$h = h_1$ and $b = b_1$

By means of common elastic line formulas we find the horizontal deflection $f_A = 0$ and $\tan \theta_A = 0$.

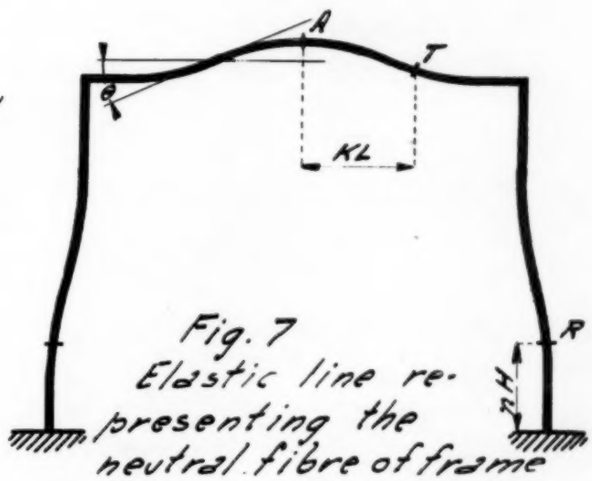


Fig. 7
Elastic line representing the neutral fibre of frame

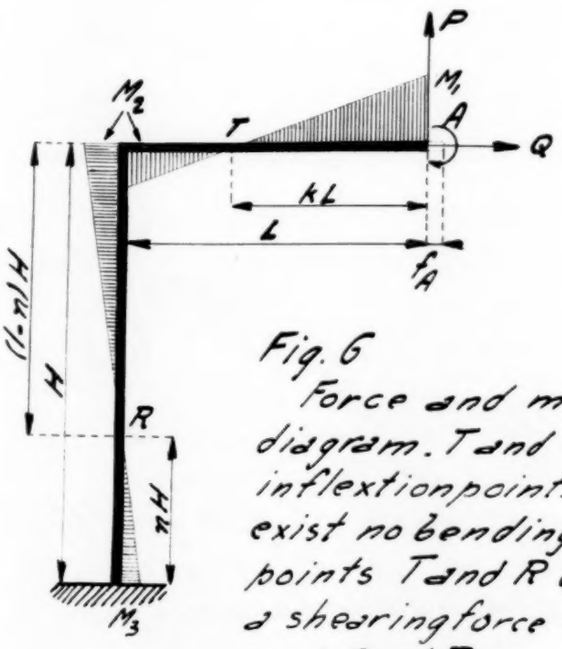


Fig. 6
Force and moment-diagram. T and R are inflextion points. There exist no bending in the points T and R but only a shearing force P at T and Q at R.

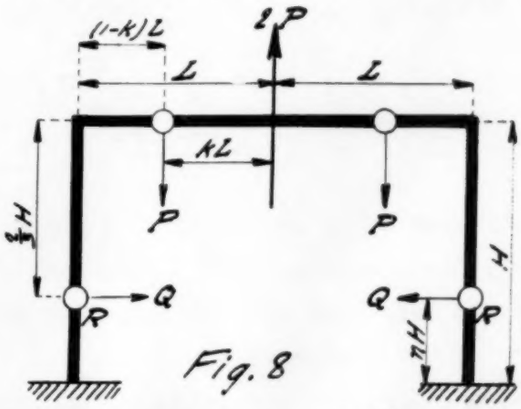


Fig. 8
Frame hinged at the inflextion points.

Plastic and Elastic Deformation.

$$f_A = 0 = \frac{(M_1 - PL)H^2}{2EI} + \frac{QH^3}{3EI} \qquad (12)$$

$$\tan \theta_A = 0 = \frac{(M_1 - PL)H}{EI} + \frac{QH^2}{2EI} + \frac{M_1 L}{EI_1} - \frac{PL^2}{2EI_1} \qquad (13)$$

in which:

M_1 = the bending moment at A to be determined

$M_2 = M_1 - PL$ " " " B " " "

$M_3 = QH + M_2$ " " " C " " "

E = modulus of elasticity

$I = I_1$ in above case $= \frac{bh^3}{12}$ = moment of inertia.

$\frac{M_1 - PL}{2EI} H^2$ = horizontal deflextion due to Moment M_2

$\frac{QH^3}{3EI}$ = " " " " " " QH

Similarly the four terms of the equation $\tan \theta_A = 0$ are the values contributed by the four moments M_1, M_2, QH and PL to $\tan \theta_A$.

PL is opposite M_1 and therefore considered negative.

The solution of the equations gives:

$$\frac{M_1}{2} + \frac{QH}{3} = \frac{PL}{2}$$

and $M_1(L+H) + \frac{1}{2} QH^2 = P\left(\frac{L^2}{2} + HL\right)$

or $$Q = \frac{(PL - M_1)3}{2H}$$

hence $$M_1 = PL \frac{2L + H}{4L + H} = PLk \qquad (14)$$

$$Q = \frac{3PL^2}{H(4L+H)} \qquad (15)$$

$$M_2 = M_1 - PL = PL(k-1) \qquad (16)$$

$$M_3 = QH + M_2 = \frac{PL}{2}(1-k) \qquad (17)$$

hence $$\frac{-M_2}{M_3} = 2 = \frac{1-n}{n} \qquad (18)$$

therefore $n = \frac{1}{3}$ and $nH = \frac{1}{3} H$ is the distance of the inflextionpoint R above the baseline of the frame.

This value of n is constant regardless of the width $2L$ of the frame.

The value of k is found from the expression:

Plastic and Elastic Deformation.

$$k = \frac{2L+H}{4L+H} \quad \text{for } I = I_1 \qquad (19)$$

we find then kL = distance of T (fig. 6) from A

for $L=H$	$k=.6$	$M_1 = +.6PL$	$M_2 = -.4PL$	$M_3 = +2PL$
" $L=10H$	$k=.51$	" $= +.51PL$	" $= -.49PL$	" $= +2.45PL$
" $L=.1H$	$k=.85$	" $= +.85PL$	" $= -.15PL$	" $= +.075PL$
" $H=$ zero	$k=.5$	" $= +1.00PL$	" $= -.5PL$	" $= +.25PL$

The application of the formula gives the maximum bendingmoment at section A-A Fig. 5

Let S = maximum allowable stress in lbs/sq"
W = sectionmodulus at A-A in inch^3

then $M_1 = WS = PLk$

or maximum allowable force

$$P = \frac{WS}{Lk}$$

Let $b = 20"$, $h = 10"$ (fig. 5), $L = H$, then we find for $L = 20"$ and $S = 20000$ lbs per sq. inch.

$$W = \frac{bh^2}{6} = \frac{2000}{6} = 333, \quad k = .6$$

and $P = \frac{333 \times 20{,}000}{20 \times .6} = \frac{333{,}000}{.6} = 555{,}000$ lbs

and $2P = 1{,}110{,}000$ lbs = total pressure.

The forgoing example is the simplest case because we took $h = h_1$ and $b = b_1$,
therefore $I = I_1$ or $\frac{bh^3}{12} = \frac{b_1 h_1^3}{12}$

In case we take different sections for the vertical and horizontal parts of the frame then the formula's become:

$$M_1 = \frac{PL(2LI + 4HI_1 - 3I_1H)}{(4HI_1 + 4LI - 3I_1H)} = \frac{PL(2LI + HI_1)}{HI_1 + 4LI} \qquad (20)$$

$$Q = \frac{(PL - M_1)3}{2H} \qquad (21)$$

From forgoing formula's it follows that the strength can be improved and a considerable

Plastic and Elastic Deformation.

amount of material can be saved by proper design of frames, rollstands or similar structures.

By application of the theory of least work, which is credited to Castigliano, we find the same result as shown in above formula's.

Applications pertaining to Fig. 5					
	I	II	III	IV	V
Given Data					
L inch	20	40	60	80	100
H "	60	100	200	80	300
b "	30	30	30	30	30
h "	15	15	15	20	20
b "	30	30	30	30	30
h "	20	20	20	25	25
$2P$ lbs	2,000,000	2,000,000	2,000,000	2,000,000	2,000,000
Calculated Data					
$I = \frac{bh^3}{12}$ inch^4	8437.5	8437.5	8437.5	20,000	20,000
$I_1 = \frac{b_1 h_1^3}{12}$ inch^4	20,000	20,000	20,000	39,062.5	39,062.5
$M_1 = \frac{PL(2LI+HI_1)}{HI_1+4LI}$ inch lbs	16,400,000	31,960,000	49,920,000	53,120,000	79,700,000
$M_2 = M_1 - PL$ " "	− 3,600,000	− 8,040,000	−10,080,000	−26,880,000	−20,300,000
$M_3 = QH + M_2$ " "	1,800,000	4,020,000	5,040,000	13,440,000	10,150,000
$Q = \frac{3(PL-M_1)}{2H}$ lbs	90,000	120,600	75,600	504,000	101,500
$K = \frac{2LI+HI_1}{HI_1+4LI}$	.820	.799	.832	.664	.797
$S_1 = \frac{M_1}{I} \times \frac{h_1}{2}$ lbs/sq.inch	8,200	15,980	24,960	16,998	25,504
$S_{2min} = \frac{M_2}{I_1} \times \frac{h_1}{2}$ " " "	1,800	4,020	5,040	8,602	6,496
$S_{2max} = \frac{M_2}{I} \times \frac{h}{2}$ " " "	3,200	7,147	8,960	13,440	10,150
$S_3 = \frac{M_3}{I} \times \frac{h}{2}$ " " "	1,600	3,573	4,480	6,720	5,075

The frame can be linked or hinged at the inflextion-points R and T without changing its strength or stability. (see fig. 8). This gives us the important equation:

$$PL(1-k) = Q(1-n)H = \frac{2}{3}QH \qquad (22)$$

which is a check on formulas 20 & 21.

SYMPOSIUM

The Theory and Practice in Connection With The Application of Brakes to Motors, For Cranes, Screwdowns, Mill Tables, etc.

PRESENTED BEFORE

ASSOCIATION OF IRON AND STEEL ELECTRICAL ENGINEERS

Pittsburgh, Pa., November 15, 1930

Some of the Points Considered Were:

(1) Whether to use shunt wound or series wound brakes on crane trolleys, when these trolleys are equipped with anti-friction bearings.

(2) When two motors are used on the Crane Bridge Drive, what is the best brake application from an operating standpoint? We do not mean by this that we want to know the best make of brake to put on there, but there is a question in some operators' minds as to whether to use one brake or two brakes. Also, there is a question as to whether two magnetic brakes are at all satisfactory.

(3) On the cranes where the operator travels with the trolley, how about the brake on the bridge motor and how should it be operated?

(4) What are the advantages or disadvantages of motor mounted brakes versus floor mounted brakes?

(5) We believe a reference to the proper and regular inspection of brakes would be a benefit. We would particularly stress the point of maintaining the proper air gap according to the manufacturer's recommendations, and there are probably a lot of other features in the different makes of brakes which should have the attention as per the manufacturer's instructions, and which do not get it.

(6) A discussion as to the merits or demerits of cast iron, cast steel, forged steel of high carbon content, or the tool steel wheels, hardened and toughened.

BRAKE APPLICATIONS IN THE STEEL INDUSTRY—
By R. M. Bayle, Steel Mill Engineer, Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

LIFE OF BRAKE LININGS AND WHEELS—
By H. E. Hodgson, Mechanical Engineer, Cutler Hammer, Inc., Milwaukee, Wis.

MAGNETICALLY-OPERATED BRAKES—
By J. H. Hall, Electrical Engineer, Electric Controller & Mfg. Co., Cleveland, Ohio.

SHUNT OR SERIES BRAKES ON ANTI-FRICTION BEARING EQUIPPED TROLLEYS—
By W. C. Raube, Industrial Engineering Dept., General Electric Company, Schenectady, N. Y.

THE THEORY AND PRACTICE IN CONNECTION WITH THE APPLICATION OF BRAKES TO CRANE MOTORS—
By R. J. Wadd, Mechanical Engineer, Harnischfeger Corp., Milwaukee, Wis.
By G. W. Baumgarten, Assistant to Electrical Engineer, Carnegie Steel Company, Duquesne, Pa.
By W. L. White, Electrical Dept., Carnegie Steel Co., Duquesne, Pa.
By A. J. Acker, Sales Engineer, Manning, Maxwell & Moore, Inc., Pittsburgh, Pa.

Discussion:

H. W. BALL, Engineer, Morgan Engineering Co., Alliance, Ohio.
JAMES FARRINGTON, Elec. Supt., Wheeling Steel Corp., Steubenville, Ohio.
F. O. SCHNURE, Elec. Supt., Bethlehem Steel Company, Sparrows Point, Md.
J. P. BIGGERT, Sales Eng., Tool Steel Gear & Pinion Company, Pittsburgh, Pa.
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R. M. BAYLE, Steel Mill Engineer, Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa.
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Brake Applications In The Steel Industry

By R M. BAYLE*

The successful retardation of a rotating force can be accomplished by a number of methods. These retardation forces can be broadly classified into two general headings, namely, those operating by the counter torque from a prime mover and those produced by frictional devices. In the realm of industrial electrical applications the magnetic brake is the most widely used frictional device where automatic operation is required. In the application of a retarding device three questions face the engineer:

1. What kind of a retarding device shall we use?
2. What form of this retarding device shall be applied?
3. What size shall it be?

This discussion confines us to modern steel mill motor applications. The first question that faces a steel mill engineer is whether or not the application requires a brake or some other means of retardation such as dynamic braking or plugging. To generalize on this question is exceedingly dangerous. There is a great deal of divergence of honest opinion as to whether or not magnetic brakes should be applied to a great number of steel mill drives. Only a few of the basic facts governing such a decision will be mentioned.

Given the choice of the above three retarding forces we can define their fields as follows:

Plugging:

This method of retardation is eminently satisfactory where an accurate stop is not required. It is also extremely fast and makes use of the prime mover itself with very few additions to the control. There is the disadvantage that it is inoperative in case the power supply fails. In processes where numerous reversals are made it is used extensively.

Dynamic Braking:

Dynamic braking is a form of counter-torque where the electric motor is converted into a generator and dissipates the stored energy of the system through a suitable loading resistance. It is most effective at the operating speed of the motor and less effective as the motor approaches a stop. Its use is confined to processes where an immediate stop is necessary upon the removal of the rotating power. This form of retardation has the disadvantage that shunt or compound motors are necessary for a simple circuit.

Magnetic Brake:

The magnetic brake in conjunction with an electric motor has the broadcast field of application. It is safe against power failure and has a fairly constant decelerating torque over the entire speed range. Its operating parts are mostly mechanical and the operating condition is, therefore, visibly apparant. It can also be used to supplement other retarding forces and requires no elaborate interlocking for such service.

As a general rule, plugging should be used for rapidly reversing cycles which do not require an accurate stop. Dynamic braking should be used where a rapid retardation upon the removal of power is required and there are no positive limitations in the accuracy of the stop. Magnetic brakes should be used where it is impossible or impractical to use an electrical method of retardation and a positive stop is necessary. Magnetic brakes should always be used where it is necessary to hold a load from rotating against external forces. A combination of dynamic braking and a magnetic brake should be used where extremely rapid deceleration is required with an accurate stop.

If our application dictates that a magnetic brake is necessary the next question is what form of magnetic brake shall be used. We have magnetic brakes suitable for direct current and magnetic brakes suitable for alternating current. The direct current brake is superior to the alternating current brake in power consumption, release characteristics and in torque per pound of weight. Steel mills use direct current in the majority of cases so these favored characteristics are available for the severest applications. However, should the application be powered with alternating current suitable brakes are available. Where exceptional performance on A.C. is required it might be well to mention that considerable progress is being made in the field of applying direct current brakes to alternating current applications. Suitable rectifiers of the copper oxide type are being manufactured which can supply rectified alternating current to the direct current brake and thus translate the desirable operating characteristics of the direct current brake to the alternating current field.

If a direct current brake is necessary we have the series and shunt types from which to choose. Here again, we have a widely diversified field of opinion. The series brake ranks high from the safety standpoint since it insures a sufficient current must be applied to the motor to hold the load before it will release. By reason of the fact that its operating current is high, ground faults that would tend to keep it open will manifest themselves more readily and operate the protective equipment of the drive. The series brake has the drawback that it requires approximately 40% of the full load current of the motor to release it and that it will hold in, only down to approximately 10% of the full load current of the motor. If the load is overhauling or fails to extremely low limits it cannot be used.

The shunt brake has the advantage that once energized it retains its release regardless of the load

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on the motor. Its disadvantages are that it requires extra control wires and usually some form of initiating device on the controller. It is susceptible to grounds on the system which cause a holding circuit that may or may not give an indication to the operator. To obtain a speedy release the coil must be designed with a low impedance which makes a protective resistance in series necessary. Thus, in deciding this application question usually one finds safety factors on one side and operating performance on the other. Since safety cannot be absolutely disregarded it is often necessary to provide some other added safety feature in order to retain the desired operating characteristics.

With the form of magnetic brake settled upon, it is necessary to decide the size of brake required for the particular application. Brakes are commonly rated on the torques they can deliver over a specified time. In other words, the coil should have the same temperature rating as the motor with which it works. Within certain other limitations a one-hour rated motor should have a one-hour rated coil. The standard ratings for series coils are ¼ hr., ½ hr., 1 hr. and continuous. Shunt coils are usually rated either as intermittent or continuous. Where the brake is connected to the line for periods of more than one-half the time a continuous coil is used.

In determining the maximum torque of the brake required the familiar formula is used:

$$\text{Torque} = \frac{5250 \times \text{H.P.}}{\text{Full load RPM.}}$$

A brake should be selected which has a torque rating equal or above the torque value obtained from this formula. With the temperature time rating of the motor and the torque value required the manufacturers data sheet will give the brake size. In case where adjustable speed motors are used the torque at the lowest operating speed should be selected.

A great number of the modern production cycles make use of intermittent motor service. A magnetic brake must dissipate its retarding force in heat. Rapid cycles therefore dictate that the brake heating capacity shall be investigated. Brake manufacturers have also listed in their data sheets the brake capacity in horsepower seconds per minute. Any new motor application requires that the power required for acceleration be taken into account as well as the power to run the apparatus once it is up to speed. In accelerating a motor drive the frictional resistance must be overcome by additional power. In braking this same power must be dissipated minus the frictional resistance which is now aiding the retardation. Such power requirements can easily be estimated or secured from actual test data on similar applications. Knowing the frequency of operation, that is—the number of times per minute the brake will operate, we have the necessary information to calculate a figure representing the horsepower seconds per minute of our application. Similar to the torque rating this value should be equal or less than the value given for the particular brake size in the manufacturers information data.

Some times the brake capacity must be increased over normal limitations by reason of peculiar safety factors. Consider the application of a brake to the motor of a blast furnace skip hoist. On the common balanced skip type the brake must be large enough to hold a loaded skip should the rope break and remove the counter-balancing effect of the empty skip. Again consider the application of a brake in an exceedingly high ambient temperature. It is natural to assume that the heat produced by the brake cannot be dissipated as easily as in normal temperatures and, hence, a larger size would be required.

Having determined on the proper brake to use the mechanical application should be such that the brake can perform its required duty with the minimum of maintenance. Brake shoes or bands have to be replaced and motor armatures have to be changed. The mechanical design of the apparatus as well as the design of the brake itself should make these changes easy. The support of the brake should be capable of taking the torque the brake will be required to exert. Not only should strength be considered but rigidity. Motor mountings are preferable because they transmit the required torque to a common source, namely, the motor feet. Some forms of brakes by reason of their design must be protected from misalignment. A motor mounting designed by the manufacturer effectively standardizes this limitation and largely removes the responsibility from the mill operating force.

To conclude, one might say the ideal magnetic brake has not yet been built. Each manufacturer stresses certain desirable characteristics and there are other desirable characteristics common to all. Regardless of the manufacturers product a much improved operation will be secured by giving due consideration to the proper application, as governed by sane engineering practice based on both the fundamental rules and good operating experience.

Life of Brake Linings and Wheels

By H. E. HODGSON*

This paper contemplates making only a brief mention of the points outlined in the notice of the meeting, other than the final point on which some detailed discussion will be made. We believe that we can contribute most to the meeting by concentrating our discussion on the life of the brake linings and brake wheels, since we have done a considerable amount of investigation in this direction. In connection with the other points which are listed in the notice of the meeting, there are many factors which can be determined only by a consideration of the particular problem at hand, or by the design of the particular piece of equipment, so that but little can be offered regarding the best brake selection until the details

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This paper contemplates making only a brief mention of the points outlined in the notice of the meeting, other than the final point on which some detailed discussion will be made. We believe that we can contribute most to the meeting by concentrating our discussion on the life of the brake linings and brake wheels, since we have done a considerable amount of investigation in this direction. In connection with the other points which are listed in the notice of the meeting, there are many factors which can be determined only by a consideration of the particular problem at hand, or by the design of the particular piece of equipment, so that but little can be offered regarding the best brake selection until the details

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of the equipment are known. We expect to hear from the users of brakes and hope to get points which will help us to build greater service into brakes and benefit industry.

In general, on trolleys equipped with anti-friction bearings, brakes are needed only to hold the trolley from creeping due to slope of the rails of the crane, or due to angle hitches. In the trolleys equipped with the ordinary sleeve bearings, there was sufficient friction so that brakes were rarely needed. With the trolleys equipped with anti-friction bearings, a brake may be needed to give controlled friction to replace the non-controlled friction which existed in the old design of trolleys with sleeve bearings. Owing to the rather small distance travelled each acceleration, it is doubtful if the motor speed will increase and motor current will drop dangerously. It is to be noted that very low torque is required by the equipment, but that a certain limiting amount of lining area is desired to provide a long lining life, and for that reason a brake which has enough lining area will have an excess in magnet capacity so that the magnet can hold against the low spring compression to such a low current value that there will be very little chance that a series brake will apply against the wheel. As a general proposition then, we favor the use of the series brake.

In general, the foot operated brake is very satisfactory on bridge drives and the magnetic brake is used only for emergency or overtravel stops. In bridge drives the accelerating and decelerating forces are limited by the traction between the wheels and the rails. Assuming that the question relative to the two motor bridge drives refers to the arrangement in which one motor is mounted on each of two through shafts, a proper sized shunt brake on each motor would be satisfactory to get maximum retardation. The brake would be held permanently released except in case of voltage failure or overtravel.

When the operator travels with the trolley so that a foot operated brake for ordinary service stops is difficult to arrange, the bridge can be controlled by the operator with shunt wound magnetic brakes. The slower timing of the shunt brake works in well for an inching type of control. The operator soon becomes adept and can bring the bridge to stop without shock.

Whether a brake should be motor mounted or pedestal mounted has been discussed for a considerable period of years, and we have no new points to present.

Brakes should be given the same inspection service as other electrical equipment.

The question of the proper wheel to use in order to get the economic installation is a question of the severity of the service to a large extent. During the past seven years we have conducted a number of tests which have indicated certain things and we are working in the direction of the indications. Our tests started with the idea of testing brake linings, but we immediately found that we could not test brake linings alone, but must always consider both braking surfaces.

The collection of data is necessarily slow since any form of expedited tests introduces factors which reduce the dependability of the results. The slowness of the tests has the bad feature that direct comparisons of linings is difficult. The lining manufacturers are constantly changing mixtures and such changes render previous test results obsolete.

Investigations relative to the wear of asbestos base friction linings have been in progress many years. Almost universally in these tests some scheme of expedited tests was used. The friction surfaces revolved continuously at a moderate speed. Unit pressures were kept low in order to keep the temperature within reason. Another way of limiting the temperature was by water-cooling the wheel. The common testing machines incorporated a wheel driven by a motor and a shoe lined with the material under test was applied against the wheel. Means were provided to impose a known load on the shoe and to measure the tangential pull produced. The wear was measured either by weighing the lining sample or by measurement of thickness, and the wear characteristics were determined by combining the energy loss in the lining with the rate of wear. These tests provided uniform heating conditions and uniform speed. There was no instantaneous temperature change, such as is produced when a lining engages at high pressure with a high speed wheel. There was always some doubt about the value of the results obtained in this manner when applying them to commercial devices.

Varying speeds as experienced in service may affect the wear. Fairly high lining pressures and wide ranges of speed variation will be experienced in service. The lining pressure during the continuous test runs varied from six to ten pounds per square inch and a slip speed of approximately 1100 feet per minute was used. In commercial brakes the unit pressures vary from 15- to 50 pounds per square inch and slip speeds vary from 6500 feet per minute to standstill. Thus if brakes were to be built conservatively enough so that at all times they would give a maximum life, braking surfaces from three to ten times the present areas would be necessary.

Various experiences with commercial apparatus indicated that the results obtained by the continuous test process using low intensity of lining pressure and slow speed, were not reliable for use in constructing commercial devices. Unsatisfactory lining life in service led to a decision to make tests under commercial conditions. The range of pressures from 10 to 70 pounds per square inch and slip speeds of 850, 1650 and 2450′ per minute were studied. This leaves a large range of conditions unexplored, but the tests are in an interesting field. Observations of friction equipment had indicated that lining life might vary according to the following relations:

1. Exponentially according to pressures.
2. Exponentially according to speeds.
3. In some non-mathematical relation to temperature.
4. In some relation to wheel material.
5. In relation to surface conditions of the wheel.

It might be well to mention that the tests did not conclusively settle the points, but did provide some indications relative to each.

To study the conditions an equipment was made up to reproduce operating conditions and record the results. A fly wheel was mounted on a shaft with a brake wheel and a motor drive. An electro-magnet-

ically-released, and spring-applied brake was used. The brake was supported on a beam which was supported on ball bearings on the shaft. The outer end of the beam was restrained by a mechanism which recorded the force at the time the brake applied and the distance of slippage between the brake lining and the brake wheel. In each cycle the fly wheel was accelerated to a predetermined speed; the motor circuit was then opened and the brake was applied. The recording instrument showed the retarding force and the number of revolutions of the brake wheel during deceleration. The shoe pressure was determined by measuring the spring compression. The number of cycles was recorded by a counter and the period of the cycle was controlled by a motor driven timer. The wear was measured by micrometer measurements taken at various marked points on the shoe. Thermometer readings on the wheel face were taken at regular intervals. The wear characteristic was calculated in foot pounds of energy required to wear away .001 inch thickness for each square inch of area.

Various samples of lining were attached to the shoes. With each sample a preliminary run was made to produce a uniform wearing surface.

Readings were taken for comparison, only after the preliminary run was completed. The test conditions were made progressively harder by increasing only one element at a time. The shoe pressure would be changed and a set of readings would be taken. The wheel speeds would then be changed and the frequency of cycles were reduced to hold the temperature unchanged. Readings taken under these conditions were expected to indicate to some extent the effect of speed upon the rate of wear. Then the frequency of cycles would be increased to determine what the effect of temperature changes would be. Tests were made on one or more samples of each of three major kinds of linings—woven, moulded, and folded and stitched.

The basic material in all cases is asbestos. In all but moulded types it is fashioned in to a thread around a brass wire. Generally some cotton is used to make the thread more easily formed. The woven linings are made so that in effect there are several layers of woven fabric tied together by occasional strands of woof which pass through all layers. The fabric is then impregnated with a binder which will serve as a matrix to cement the strands and fibres together. Tar, asphalt, sugar, silica, rubber and various polymerizing oils are or have been used. Then heat and pressure are applied to cure the binder and reduce the lining to the final shape and condition. In this process the thickness which will give longest life per unit of material may be limited by the wearing process, the penetrating process of the binder, or by the press capacity. The woven lining is reasonably flexible and can easily be formed to suit wheel surfaces. Its surface is rather rough and the high spots must be worn down before maximum friction is delivered. In the moulded type the asbestos fibres are mixed with a binder and other constituents. Some times short pieces of brass wire are used to re-enforce the product. The mixture is then formed in moulds to the desired shapes. The moulds are customarily heated to cure the binder. Moulded linings are hard and dense. They are not so easily fastened to the carrying surface as either of the other types and may break. They are uniform in thickness and the surface is smooth. They must be formed to the shape in which they will be used.

In the folded and stitched type of lining the strands are loosely wooven into a rough open cloth, which is filled with the binder by the frictioning process. In the frictioning process the cloth is passed through between metal rolls. The binder material is fed into the bite of the rolls and adheres to them to some extent, so that it is pulled through with the cloth and forced into all spaces in the cloth. Usually only one surface is filled at a time and two passes are required to cover both surfaces. The friction cloth is folded together in the necessary number of sheets to build up to the desired thickness. They are then loosely stitched to make further handling during the subsequent operations easier. The stacks are then subjected to heat and pressure for curing and producing a uniform thickness. The folded and stitched linings are reasonably flexible and will not break while being riveted to the carrying surfaces. The surface is moderately smooth.

The fundamental requirement in all types is that the binder must permeate the fabric and form a matrix to support the relatively weak fibres of asbestos. The binder must stand the temperature which develops when the wheel slides against the friction lining under pressure and must remain mechanically strong enough so that it will support the asbestos. When linings are worked too hard the wear rate is multiplied in all types. There is a critical temperature which if exceeded causes rapid failure of the lining. It does not appear that the wear rate is especially affected at any value below the critical temperature of the particular lining. When the critical temperature is exceeded in the woven lining, the woof cuts away and the binder either is not strong enough to hold the pieces or it vaporizes out and strands of the warp fly out along the short loops of woof.

The moulded type scuffs away.

The folded and stitched type separates between layers and partial layers flake off.

The problem is to know what is the limiting severity of service to give economic results.

The adhesion or co-efficient of friction can be regulated by the manufacturer and appears mainly to be a function of the binder in combination with the asbestos. The co-efficient is not exact and will vary in a given sample, over a range of 1.6:1. The factor depends quite largely on the condition of the metal surface which opposes the lining, as well as upon the lining itself. Roughly, the life is progressively shorter as the co-efficient of friction increases. There are temperature effects, oil effects and water effects which produce variations in results. There is a light service condition where a film of metal oxide will form on the lining face practically eliminating the lining wear and increasing the frictional co-efficient to as high as .8. This can be obtained with almost any type of friction material, if the right conditions are provided, but the conditions represent such low energy consumption per unit of area that it is not practical to embody them in industrial apparatus. We have previously pointed out that it

Forms of Cast Iron	Heat Treatment	Scleroscope	Average Lining Wear Characteristic	Coefficient of Friction	Wheel Notes
Semi-Steel Casting	None	27-28	30,900	.41	Moderate wear and ridges appeared under severe conditions.
Nickel Iron Casting	None	36-38	245,000	.42	Some wear occurred, but it was uniform and a high polish was maintained.
Graphitic Steel Casting	None		294,000	.40	Moderate wear but uniform.
Steel Wheels					Wheel scored badly under moderately hard service conditions.
.20-.30 C Casting	Annealed	About 20	262,000 22,100	.46	The low value of lining wear characteristic was obtained after wheel scoring started.
.45 C Casting	Annealed	About 25	176,000 20,600	.6	The wheel scored badly under severe service conditions and the lining characteristic dropped to 296,000.
.65 Forged	As forged	38-43	399,000	.58	Severe scoring developed at moderate severity of service. No lining wear characteristic was determined after scoring started.
.45 Carbon Casting	Quenched in oil	40	420,000	.53	No scoring occurred, but the greatest severity of test conditions were not imposed.
Mild Steel Fabricated	Carburized and hardened	60-75	1,250,000 578,000	.49	No scoring resulted under the most severe test conditions.

All tests were conducted with a representative "folded and stitched" type of lining. The lining wear characteristic was the average of several determinations under varying severity of conditions. The wear characteristic is the number of foot pounds of energy to wear away .001 inch from one square inch of lining area. The highest value obtained was 1,250,000 with light service conditions on the case hardened steel wheel.

The coefficient of friction was amply high with all types of wheels. It varied over a range with each type of wheel under different conditions.

would require from three to ten times the present accepted friction area to produce these conditions.

COMPARATIVE RESULTS WITH LININGS

Type of Lining	Coeff. of Friction	Lining Wear Characteristic Range	In order of Wear effect on Wheels	Wheels with which tests were made
Woven	.36-.75	9000 to 100000	Hard	Steel castings untreated and Steel castings — hardened.
Moulded	.3 to .6	43000 73000	Harder	.20-.30 C. casting untreated.
Folded & Stitched	.36-.75	22000 1,250,000	Hardest	All wheels in list and all service conditions.

Brake wheel materials must be strong and flexible, but must break sharply if their strength limit is exceeded. In other words, it is desirable to have the elastic limit and the ultimate strength high and close together.

In operation brake wheels heat up unequally. The source of heat is between the lining and the wheel and the heat flows through the wheel to meet a cooling current of air at the inside of the wheel; consequently the wheel expands unequally in proportion to the inequality of temperature in the wheel. Elasticity is required to stand the inequality of expansion The particles in the wheel should retain their strength up to fairly high temperatures, and when their strength is exceeded should break away in infinitesmal grains and not foul the lining. Cast iron wheels were tested in three forms—semi-steel, nickel cast iron and a so-called graphitic cast steel which is very similar in appearance to cast iron, but can be hardened. Steel wheels were tested in 20-point carbon casting and 45-point carbon casting without heat treatment, low carbon with some manganese casting, 5-point carbon forged wheel, 45-point carbon casting hardened, 60 to 75-point carbon rolled steel plate formed for the rim and hardened, untreated alloy steel, and mild steel carburized and case hardened.

With low pressures and low energy consumption, the wear characteristics go very high, for example, a half inch wide leather belt with a half wrap on a wheel running at 1200 RPM with a slip speed of 294′ per minute and a 10 oz. pull on the leather belt operated continuously for three months with no measurable wear of the leather. On another condition similar to this, a 1/32″ thick band operated a year at 200 RPM, equivalent to 100′ per minute slip speed, with no apparent wear of the band. It will be noted from a study of the tabulation that the wear characteristic values have a ratio as high at 60:1, depending upon the brake wheel condition and the temperature conditions. The range of a single lining on a single type of wheel is approximately 5:1.

The general indicated conclusions are that cast iron wheels are not elastic enough for wide ranges of temperature. Surface cracks develop and under extreme conditions cracks will progress through the rim, due to the inequality of stresses on account of the varying expansion in the rims. In ordinarily mild service the wheel surface polishes and produces very good conditions for long life of wheel and lining. Other features are that in order to get the same rigidity, the weight must be increased approximately 50%. This adds to the fly wheel effect and is undesirable. Due to the lower breaking strength the permissible speed of rotation is less for the cast iron wheel than for the steel wheels.

Steel, whether of low or high carbon content, without heat treatment, under easy service conditions with light pressure and low speeds will ordinarily polish and produce long wheel and lining life. There is a gradual improvement in wheel performance as the carbon content increases, but it was

indicated from the tests that with pressure in the range from 30 to 50 pounds per square inch and normal velocities representing fair severity of service, steel without heat treatment, tends to drag and particles bed into the lining. Then the scoring of the wheel progresses rapidly.

Hardened steel, even though the hardness does not exceed 40 scleroscope, apparently produces better lining life than cast iron in the ratio of about 4:3, without danger of cracking. The cost of the hardened, or heat-treated wheel, is certain to be somewhat above that for a casting, since the wheel must be machined close to finished size before it is heat-treated, and then it must finally be finished after heat treatment, so that two sets of operations are required in the manufacture of the wheel instead of a single set which is needed for the untreated wheel. Also the final grinding operation on the hardened wheel surface requires somewhat longer than a machined finish on an untreated wheel. The heat treatment itself adds something to the cost. However, in view of the apparent economies to be gained from the increased length of lining and wheel life, it appears that on a majority of installations it would be an economical plan to buy the hardened steel wheels.

To get the greatest uniformity of wearing surface, forged material is desirable. Also to keep the fly wheel effect to a minimum the fabricated wheel is best since the rim and web thicknesses can easily be proportioned to the force requirements. In castings, the thickness must sometimes, in order to get a sound casting, be heavier than would be necessary for strength.

Finally, then, cast iron will be satisfactory with any lining for light service and limited speeds; untreated cast steel will be satisfactory with some linings for light service and fairly high speeds; hardened steel will be satisfactory with any lining for all service conditions. Lining life varies over a wide range, depending more on wheel condition than on severity of service, except that severity of service finally affects the wheel condition.

Magnetically-Operated Brakes

By J. H. HALL*

The development of the various forms of magnetically-operated brakes followed the application of electric motors to elevators and hoisting machinery in the early replacement of steam and hydraulic drives. For any drive in which the load tends to overhaul the motor, some form of brake is necessary to slow down the load when power is cut off from the motor.

The early brakes were shunt wound, and were designed to set when power was applied to the solenoid or magnet. Later designs included the spring or gravity set type, now in general use, having either shunt or series windings. During this discussion, the terms "magnetic brake" and "magnetically-operated brake" are intended to include all types of brake which are spring or gravity set, and which are released by the energization of a solenoid, an electro-magnet, or an electric motor.

On modern direct current hoists, a large percentage of the retardation of the motor and load is effected by dynamic braking and the magnetic brake functions almost entirely as a holding brake. The same is true of high speed mill auxiliaries, such as screwdowns, side guards and manipulators when magnetic brakes are used on these motions. The reason for this is that a direct current motor can be readily controlled by dynamic braking to retard itself and the connected load, the braking being accomplished without frictional devices and without power expense.

Due to the difficulty of securing dynamic braking on alternating current motors, magnetic brakes are required to slow down and hold the load for the hoist motion on practically all alternating current cranes.

For rotating, reciprocating and horizontal travel machines, there are, in general, six methods of stopping when power is cut off from the motor:

(1) Allow the machine to drift to rest.
(2) Apply regenerative braking; returning power to the line.
(3) Apply dynamic braking; taking no power from the line.
(4) Apply reverse power; taking power from the line.
(5) Apply air, hydraulic, or hand (foot) operated brake.
(6) Apply magnetically-operated brake.

In the interest of safety and high speed operation, it is frequently desirable to use two or more of the above methods of braking.

Should Magnetic Brakes Be Used On Trolleys Equipped With Roller Bearings?

Before roller bearings came into use, reverse power or plugging was used for slow down on nearly all crane trolleys. The friction of the sleeve bearings and gearing was sufficient to allow drifting to rest in case of power failure without the possibility of serious damage. Notable exceptions to this are the heavy trolleys of ore bridges and ore unloaders, on which magnetic brakes are provided for slow down and as a safety measure against power failure. They serve also as holding brakes against wind pressures and against drifting when tracks are not level.

Trolleys, as well as bridges, equipped with roller bearings, are motored from the standpoint of acceleration. The rolling friction is so small that after acceleration the machine is over-motored. Excessive speeds are obtained, which run as high as 300% of full load motor speed.

Plugging a series motor at 300% full load speed will produce with standard controllers, a retarding torque sufficient to skid the track wheels, and the current values involved will frequently injure the motor commutator. This can be overcome by pro-

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viding suitable resistors in the controller during the plugging operation.

In some industrial plants, roller bearing trolleys have been equipped with a series low torque magnetic brake on the trolley motor to prevent the trolley rolling toward the end of the bridge girders, due to the camber in the girders. On account of the low current taken by the motor, it is necessary to supply these brakes with special coils having a greater number of turns than the standard coils for the same size motor. This is possible if the service on the trolley motor and the brake is not too severe.

On industrial and steel mill crane trolleys, except the larger ones above mentioned, magnetic brakes which set at the off position of the controller are objectionable for two reasons: (1) The inability to drift, and (2) the wear on brake linings and the maintenance required. With properly designed controllers, plugging to stop is more satisfactory from the viewpoint of upkeep and the accuracy of operation.

It is questionable whether series brakes can be used on all roller bearing trolleys, because the low value of current taken by a series motor at high speeds will not maintain the brake released. Moreover, shunt brakes de-energized at the off point of the controller are slower in releasing and tend to slow up the operation of the crane. As a measure of safety to prevent drifting into the end stops on failure of power, continuously energized shunt brakes are in some cases advisable on roller bearing trolleys. With this exception, magnetic brakes are considered unnecessary on trolleys equipped with roller bearings.

The Application of Magnetic Brakes On Two-Motor Bridge Drives

Bridge drives having two motors involve the use of two bridge line shafts, each driven by one motor, thus one motor is geared to one-half the number of driving track wheels.

On account of the simpler mechanical details crane builders prefer to provide but one foot brake on two-motor bridges. This is mounted on the bridge line shaft motor most convenient to the operator's cage. With this arrangement, the retarding effect of the brake is only one-half the accelerating effect of the two motors. This is sufficient, however, if slow down is effected by plugging or reverse power, and the foot brake is to serve only as a holding brake. A brake connected to only one bridge line shaft will tend to skid the track wheels to which it is connected, particularly if quick stops are attempted with a retardation effect equivalent to the accelerating effect. It is general practice on high speed steel mill cranes to retard the bridge motion by reverse power. In some industrial plants plugging is undesirable, slow down being effected by means of brakes. If, in addition to the foot brake, magnetic brakes are considered necessary on heavy industrial cranes for slow down, there should be a magnetic brake for each motor in order to secure equal retardation on all driven track wheels, thus making the slow down time comparable with the acceleration time. It is no more difficult to adjust two well designed magnetic brakes to deliver equal retarding torques than it is to adjust four-wheel brakes on an automobile. There is no complication in the control of two brakes on a two-motor bridge drive, particularly, if a duplex magnetic controller is used.

The conclusions are, therefore, that it is best to use reverse power for slow down on steel mill two-motor bridge drives and to use a foot-operated holding brake which may be locked for holding the crane when not in use. Heavy industrial cranes of this type may require magnetic brakes for slow down, in which case two should be used. A foot brake for parking the crane is also necessary.

Bridge Motor Brakes On Man Trolleys

With the exception of Ore Bridges and Ore Unloaders, cranes on which the operator travels with the trolley, invariably depend upon reverse power for retardation of the bridge motion. Soaking pit cranes, stripper cranes and charging machines are fast operating and require accurate control which can best be secured by direct control of the motor. If series brakes were used, the ability to drift would be necessarily surrendered. Shunt brakes which set at the off position of the controller to effect slow down will delay acceleration on account of their relative slowness in releasing. This delay is in the nature of one or two seconds for each start. The only logical use of a magnetic brake on the bridge motion of a mill type man trolley crane is a shunt brake floating on the line to stop the bridge in the event of failure of power. This is an investment in the interest of safety to workmen and machinery.

Motor Mounted Brakes Vs. Floor Mounted Brakes

The principal advantage of a motor mounted brake is the saving in the mounting supports, or brackets. This saving is somewhat offset by the extra cost of the motor. Another advantage is that the brake can be economically designed for that particular motor size.

A third advantage is that the motor and brake can be tested at the same time by the same manufacturer with no misunderstanding as to their relative ratings.

The disadvantages are:

(1) Inability to use undersize or oversize brakes on a given motor frame.

(2) Inability to readily substitute any other form or make of brake on the motor in case of emergency, or in case of the desirability of making a permanent change. If a change **is** made there is involved the added cost of a mounting support, or bracket.

(3) Some types of motor-mounted brakes prevent the change of a motor armature without dismounting the brake.

(4) An accident damaging the brake support might also damage the motor frame, requiring a complete motor frame replacement.

(5) Spare brakes are not interchangeable with floor mounted brakes of the same make, on some types of motor mounted brakes.

The disadvantages of motor mounted brakes appear to predominate.

Inspection and Maintenance

The great loss of life due to the failure of automobile brakes has resulted in stringent laws in certain cities and states requiring automobile brakes

to be in such a condition that the car can be stopped within a given distance. I know of no such state laws regarding crane brakes. However, the safety committees, in their attempt to conserve life and property can provide for proper brake inspection and maintenance.

Magnetic brakes function due to the friction between two or more surfaces, and wear is inevitable. A periodic inspection of all brakes should be made, the period depending upon the severity of the service on the brake. Wear on the friction surfaces increases the operating air gap, and in most all types of brakes, changes also the torque adjustment. The brake shoes, or brake bands, should be adjusted so that they do not ride on the wheel when the brake is released, and the smallest air gap should be maintained consistent with free operation. Too large an air gap allows the brake to set with a shock to the brake and connected apparatus. It reduces the torque so that the brake may fail to hold the maximum load. It requires more current to release, making it necessary to advance the controller beyond the first point to start the motor. EC&M brakes are adjusted for about 1/32" clearance between the wheel and shoes and adjusted for torque values based upon the best information obtainable from the customer for that particular brake application. When the brake is installed, the torque setting should be readjusted to stop and hold the maximum load to be encountered.

Inspection at least once a week should be made on all brakes with particular attention to the following:

(1) Equalization of clearance between the wheel and friction members individually and laterally.

(2) Air gap adjustment, to provide the minimum clearance between wheel and friction linings.

(3) Tightening of bolts in brake and motor bases.

(4) Examination of leads, and tests made for grounds.

To maintain the same progress that has been made in the design of cranes during the past ten years, certain improvements have been demanded in the design of magnetic brakes. The advent of roller bearings on cranes has put a greater burden on the brake for stopping and a more rugged brake design is required, with the possibility of less maintenance cost. To make inspection easier and maintenance less, the following features should characterize a well designed brake:

(1) **Brake Linings.** These should be of the most durable slow wearing material with provision for long life without replacements. The tendency in automobile practice, which is acknowledged to be the most exacting, is toward the use of molded linings; these replacing the fabric materials previously used. Heavy molded material with deep recesses for rivet heads, provide the longest service with the least maintenance.

(2) **Moving Parts.** These should have bearings with hardened steel bushings and pins where needed to prevent wear and maintaining the brake parts in proper alignment.

(3) **Magnet Coil.** This should be liberally designed and properly insulated to prevent overheating for the work it has to do. Leads and terminals should be protected from the possibility of short circuit or ground.

(4) **Mounting.** Brakes are usually designed with the understanding that they will be mounted accurately with respect to the center line of the motor. Due to possible inaccuracies in the design of the brake support, provision should be made in the brake design to permit the brake to be mounted off center from 1/8" to 1/4", both horizontally and vertically without interfering with the operating characteristics of the brake.

(5) **Changing Motor Armatures.** The most desirable design of brake will permit the removal and replacement of the motor armature with the brake wheel attached without the removal of any bolts, rods, or other parts of the brake and without disturbing any of the adjustments. A suitable device should be provided for manually releasing the friction members from the wheel. This will also provide for moving any of the rotating members of the crane for inspection or repair without energizing the brake coil with electric power.

(6) **Adjustments.** These should be as few as possible, accessible; their operation easily understood and a change in adjustment readily made. They consist of:

(a) Air gap adjustment to take up for wear on friction linings.

(b) Torque adjustment made once for any given load.

(c) Equalization of clearance between friction members on opposite sides of the wheel. There should be only one adjusting device for this and the adjustment made once when the brake is installed.

(d) Adjustment to clear all parts of each friction member from the wheel.

(7) **Brake Wheels.** It has proven by test and in actual service that cast iron is the best material for brake wheels working against friction linings. This conforms to the recommendations made by the manufacturers of friction linings. Cast iron has a coefficient of friction greater than steel, and probably due to the graphitic content, cast iron wheels operate with less wear on the wheel and maintain smoother surfaces without cutting or grooving. Some of the Safety Committees have rules against cast iron wheels due to the possibility of their breaking by centrifugal force, but it is considered that cast iron wheels are safe and are the most suitable for small wheels and for moderate speed large wheels.

Forged, rolled, or welded steel wheels have greater strength but with a smaller coefficient of friction. They are considered most satisfactory for all wheel sizes. Cast steel wheels are not so satisfactory on account of internal blow holes which do not show up on machining and which may develop to cause a failure in service.

Tool steel wheels, hardened and toughened, are the safest from the standpoint of breakage, but they have a smaller coefficient of friction than the forged steel wheel, probably because the surface attains a higher polish. Although the cast iron wheel has a smaller cost the tool steel wheels are considered worth the extra investment for the longer service obtainable.

Shunt or Series Brakes on Anti-friction Bearing Equipped Trolleys?

By W. C. RAUBE*

With the use of anti-friction bearings on crane trolleys, some new problems arise in connection with the proper application of electrical equipment. I believe it has been customary practice on sleeve bearing trolleys to use a series wound motor without magnetic brake but with reversing control that can be plugged for stopping the trolley. On such equipments, the propelling torque is an appreciable amount and loads the driving motor sufficiently so as to hold down the speed to moderate values. Reversing controls for such drives have been standardized for years providing proper resistance to give good operating characteristics including satisfactory plugging to decelerate the trolley.

With the use of anti-friction bearings, the propelling torque is considerably reduced from that required on sleeve bearing trolleys and if the same gearing is used as would be used for a sleeve bearing trolley, it is evident that the motor will be more lightly loaded and thus accelerate to higher speeds due to the rising speed characteristic at light loads for series motors. Under such conditions then if we were to let the trolley coast, it would coast longer than with a sleeve bearing trolley, first due to the fact that the trolley and revolving parts had been accelerated to a higher speed before coasting started, and second because the running friction is lower and therefore offers less resistance to free running.

To deal with such a set-up, we could of course arrange the control so as to apply a heavier plugging torque to decelerate the trolley more rapidly, but the difficulty with this is that the plugging torque might be too severe and produce undue swinging of the load. As a matter of fact, the plugging torque produced with the standard resistor layout commonly used on sleeve bearing cranes automatically gives higher plugging torque on anti-friction bearing trolleys on account of the higher speed at which the equipment is being plugged, and therefore it may very likely become desirable to use a little higher resistance in the first step of starting resistance in order to limit the plugging effect and thereby prevent undue swinging of the load. It seems to me, therefore, that with such resistor layout, it should be possible to handle the trolley just as well as is being done on the sleeve bearing equipped trolley and standard resistor layout and without the use of a magnetic brake.

Nevertheless, there is a greater possibility to damage the equipment by over-coasting in the case of the anti-friction trolley, and it would seem desirable therefore to consider a magnetic brake which would automatically apply in case of power failure and limit the coasting distance. Such brake could be a partial torque brake in order to give less abrupt stopping.

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Whether this brake should be shunt wound or series wound must be determined by the operating conditions. Other things being equal, preference should of course be given to the brake which is the simplest and which introduces the fewest complications.

Under normal operating conditions, deceleration of the trolley is accomplished by plugging the control, and it is my opinion that the magnetic brake should not necessarily set each time that the controller is notched to the off position. The function of the magnetic brake should be to serve as a protective device preventing damage from over-coasting of trolley and therefore should stop the trolley in case of failure of power or in case of an emergency. In order to meet these conditions, a shunt wound brake seems to be the logical answer. This type of brake would automatically set in case of failure of power and could be manually applied to give emergency braking by tripping the emergency control switch used with protective panel, and which in turn cuts off power from the entire crane.

An advantage of the shunt wound brake over the series wound brake for this service is that the brake will not drag due to low line currents and low line currents are most certain to occur at times when the controller is notched from the full running point to an intermediate point.

The use of a shunt type brake will necessitate the addition of an additional trolley wire.

I wish to digress long enough at this point to discuss one angle of the anti-friction bearing trolley (or bridge for that matter) which I believe should probably be given more attention not only by electrical manufacturers but also by the ultimate user and builder of the crane than has been in the past. I pointed out the fact that the propelling torque on a trolley with anti-friction bearings was less than that on a sleeve bearing trolley and further that the motor would operate at a higher speed under such conditions. Furthermore, static friction will be reduced on an anti-friction bearing trolley, and theoretically at least we should be able to use a smaller motor than was formerly used for the same trolley when equipped with sleeve bearings. The next point is that whatever motor is applied to the job, the gearing should be properly arranged so as to give the correct trolley speed. Let us take a little more concrete case. We will assume that for a certain sleeve bearing equipped trolley, a given motor and gearing has been used and that now we want a trolley to operate under the same conditions, but that the trolley is to be fitted with anti-friction bearings. The first point to be considered in this case would be whether, with the reduced torque requirements, a smaller motor might be used to meet the service. If the indications are that the next smaller rating will provide ample torque, it should be considered at a saving in first cost over the larger motor.

Should it happen that the next smaller rating would not provide enough torque margin to make it advisable to use such motor, then use the larger frame size but be sure to properly arrange the gearing. In order to do this, the straight propelling horsepower should be determined and then refer to the characteristic curves for the motor under consideration to find the motor speed corresponding to the horsepower load, and establish the gearing so that with the motor operating at that speed, the trolley will move at the desired speed in ft. per min.

These same precautions are of equal if not greater importance on the bridge drive. It has been my experience that bridge and trolley speeds on cranes with anti-friction bearings very often are considerably in excess of rated values, thus requiring special adjustments in control resistors.

In case there are good reasons for using the same motor size and gearing for anti-friction bearing trolleys as has been used on sleeve bearing trolleys, the use of compound wound motors instead of series motors will hold down the free running speeds and may, therefore, be considered as an alternative arrangement for such trolley and bridge drives.

Best Brake Application on Two-Motor Bridge Drive

Just what is the best brake application from an operating standpoint on a two-motor bridge drive is a difficult question to answer. It is my opinion that operator's experience will be more valuable in arriving at the answer.

Considering the fixed cage type crane, I doubt whether magnetic brakes need to be used at all if a foot brake is supplied. The advantage of the foot brake is that the operator can apply braking effort in the degree required to meet operating conditions. If magnetic brakes were used, we would have no graduation of braking effort and it would therefore make it difficult for an operator to efficiently manipulate the crane.

As a matter of safety, it might be desirable to fit such foot operated mechanical brake with an electrically operated latch which would allow the brake to set in case of failure of power. This feature hardly seems necessary, however, as it is logical to assume that the bridge operator should be able to manually apply the brake and to the degree required to meet operating conditions.

I have not had any operating experience to back up any statements regarding the use of one or two magnetic brakes on bridge drives equipped with magnetic brakes, but it would seem to me that the use of two brakes would have some advantages over a single brake.

Bridge Motor Brake for Man Trolley Bridges

On cranes where the operator rides with the trolley, the use of a foot operated mechanical brake is not so easily accomplished. A very satisfactory arrangement can be worked out, however, by providing a spring set brake which can be electrically released by means of a General Electric variable pressure operator. Normally, the operator exerts maximum pressure to keep the brake released and when braking effort is required, the crane operator can press down on a foot operated switch to insert resistance in series with the operator, thereby reducing the pressure and permitting the spring to produce braking effort. Resistance can be inserted in series with the operator in a number of steps, thereby giving variable degrees of braking and duplicating to a certain extent the flexibiilty obtainable on the ordinary foot operated brake.

The above arrangement incorporates another feature, namely that in case of power failure the operator becomes entirely de-energized and automatically allows the bridge brake to be applied. During such automatic application of the brake, the inherent characteristics of the operator are such that the pressure from the operator will not unbuild instantly, meaning therefore that the braking effort will be gradually applied.

Motor Mounted Vs. Floor Mounted Brakes

Magnetic brakes may be either floor or motor mounted. The advantages and disadvantages for floor and motor mounted brakes are as follows:

Floor mounted

Advantages

1) Lower first cost.
2) If set on sturdy base, will give a more rigid installation than motor mounted type.

Disadvantages

1) Requires sturdy base.
2) Requires more skill in lining up.
3) Unless clamped very securely, can get out of alignment more easily than motor mounted type.

Motor mounted

Advantages

1) Requires no special foundation.
2) Automatically lined up with motor.
3) Alignment with motor maintained.
4) Can be handled as a unit with the motor.

Disadvantages

1) More expensive.
2) Subjects motor frame to extra strains.

Inspection and Adjustment of Brakes

As brakes are subject to more or less wear, adjustments must be made from time to time in order to keep the brakes in the best operating condition. Occasionally the replacement of brake linings or refacing of brake wheel may be required. Because of these facts, periodic inspections of the brakes should be made, and whenever such inspection reveals that the brake linings have worn to the point where the brake may soon improperly perform its duty, adjustments should be made at once in order to avoid a brake failure with its resulting consequences.

When a brake is first set up it should of course be properly lined up and securely fastened in that position. The necessary adjustments should then be made on the brake and they usually consist of:

1) Air gap adjustment.
2) Spring adjustment for torque.
3) Adjustment for clearance at back stop.

Instructions as outlined by the brake manufacturer should be followed in making these adjustments. On brakes which are not subjected to severe duty the wear of brake lining and wheel will be reduced and adjustments need to be made less frequently. The point is, however, that as the brakes are used they are subjected to wear, only slowly on

infrequent service and more rapidly on severe duty and should be given a periodic inspection. At such periodic inspections the general condition of the brake should also be noted and working parts oiled as called for by manufacturers' instructions.

Kind of Material to be Used for Brake Wheels

I believe that the experience of superintendents in charge of cranes will be very valuable in determining the kind of material to be used in brake wheels. I will, however, attempt to outline the relative merits of different types of brake wheels as I know them. Cast iron brake wheels have been used quite extensively in the past especially on some of the smaller brake sizes. This type of brake wheel is inexpensive and gives fairly good wearing qualities, but has the drawback that the wheels are subject to breakage. The cast steel wheel on the other hand is less subject to breakage, but is more expensive. The wearing quality of the cast steel wheel is sometimes uncertain. A cast steel wheel will go into service and appear to give very satisfactory service but may later develop a streak of cutting which appears to start with the embedding of steel in the face of brake lining. More recent practice has been to use rolled steel of high carbon content. These wheels appear to have much better wearing qualities than cast steel wheels. One user of these wheels estimates about four times the life for a rolled steel wheel of high carbon content as compared with cast steel wheels. The cost of these wheels is probably no greater than that for cast steel wheels.

A fourth type of wheel using tool steel which is heat treated and toughened has been used, but I only have limited knowledge of this type of wheel. The cost of such wheels is undoubtedly higher than rolled steel wheels, and it is claimed that they will give longer service than the rolled steel wheel. I trust that the discussion will bring forth more data regarding this type of wheel.

Aside from the kind of material used in brake wheels another question of vital importance should be considered. A great many of the applications on which brakes are used in steel mills call for fast reversing service. It is important therefore, in the interest of fast acceleration and deceleration to keep the inertia in the revolving parts down as much as possible. This consideration was admirably taken care of in the design of the General Electric MD-400 line of motors, and it is my desire to know whether the users of brakes have any fixed ideas regarding weights or dimensional design which would have a bearing of the WR^2 of brake wheels. As the weight in the rim of the brake wheel has a much greater influence on the WR^2 of brake wheel than corresponding weight at or near the hub it is evident that the greatest showing can be made when dealing with the rim of the wheel. I hope, therefore, that the discussion will touch upon this point and outline limitations which would directly or indirectly, have a bearing on the WR^2 of brake wheels.

The Theory and Practice in Connection With The Application of Brakes to Crane Motors

By R. J. Wadd, Mechanical Engineer, Harnischfeger Corporation, Milwaukee, Wis.
By G. W. Baumgarten, Assistant to Electrical Engineer, Carnegie Steel Co., Duquesne, Pa.
By L. W. White, Electrical Department, Carnegie Steel Company, Duquesne, Pa.
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By R. J. WADD*

(1) The use of shunt wound or series wound brakes. The use of a shunt wound brake for a direct current circuit is objectionable on account of the inherent points of failure of this type of winding. For desirable operation this brake should not be over a 30 or 40% motor torque brake and it is desirable even then to allow a drift point on the controller.

(2) In case of a double bridge drive, that is a drive on both main girders with a cage attached to bridge, it is desirable to equip one drive with foot brake and a magnetic brake on the other drive with a torque rating of from 60 to 100% motor torque on 4-wheel cranes. On 8-wheel cranes it is desirable to have a foot brake and a 100% magnetic torque brake. The use of two full torque magnetic brakes is very objectionable for operation even with a drift point on account of the inability of the operator to slow down at all times.

(3) On cranes where the operator's cab is attached to trolley the same objection exists to magnetic type brakes as with the cage attached to bridge. A mechanical operation is decidedly preferable and can be made as near 100% efficient as the application of the foot brake with the cage attached to bridge. As an additional safety measure it is well to not overlook the fact that a dynamic braking connection for either the bridge or trolley motion can be utilized as an operating brake independently or with the foot brake.

(4) The disadvantage of a motor mounted brake is the inability to maintain a rigid connection which the floor mounted brake can provide. It is pos-

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sible that floor mounted brakes can be arranged so as to be undisturbed in the replacement of armatures, which is not true with the motor mounted type. There is one additional objection to the motor mounted brake, especially in the case of improper adjustment or neglect, and that is, the impact is transmitted directly to the motor frame and the attachment bracket, which can be more adequately supported and guarded against in the floor mounted type. Undoubtedly proper care and adjustment on the brakes would eliminate this weakness.

(5) The crane manufacturer covering other fields besides the steel mills has considerable use for the chilled iron wheel. I doubt if anybody will uphold the use of the cast iron wheel on any crane as an economic investment over a forged steel wheel either in the rim toughened or heat treated type. It is the writer's opinion, however, that the ordinary forged wheel without heat treatment is the cause of considerable complaint on flange and rail wear. The use of the taper tread does act as a partial cure for the soft wheel and is a desirable asset for those wheels with a toughened rim. In the writer's opinion the use of a tapered tread is a debatable point on either a chilled iron or a pack hardened wheel on account of the increased rail loading. The use of larger diameter wheels or a reduced loading on the conventional diameter would partially eliminate this latter objection.

By G. W. BAUMGARTEN*

(1) Trolleys equipped with anti-friction bearings are generally difficult to handle if there is any swinging of the load. This situation becomes very serious if the crane is a ladle crane handling molten metal where any spillage of metal will endanger workmen's lives. We ran into this difficulty on our ladle crane trolleys and decided to equip the trolleys with magnetic brakes. It was apparent that a shunt wound brake could be connected so that it would set either on failure of power, in which event it would be of no use during normal operation, or would set at the off point of the controller which might interfere somewhat with ordinary operations, but would prevent drifting of the trolley. A more serious objection is the necessity of installing two extra collector bars.

A series wound brake if applied according to the motor size might set during operation due to the high starting and low running current required for the trolley. To determine these limits we obtained a graphic current record on these trolleys and after consulting the brake manufacturer found that we could use a brake coil normally suitable for a 15 HP. motor on this 33 HP. job.

These brakes have been in service over two years and the installation has been satisfactory in every respect.

(2) On a duplex bridge drive, it may seem logical to argue that if two motors are required to drive a bridge that two brakes should be required to stop it, but this thought neglects the possibility of plugging the motors to stop the bridge. If this is done, a brake is only needed to stop the drift, or to "spot" the bridge. One brake is sufficient for this type of service.

Another thought is that with anti-friction bearing cranes and high speed bridges there is a possibility of losing contact on the third rail or of breaking or dislocating a collector shoe, when near the end of the runway. Under these conditions, it is impossible to plug the motors and the bridge brake must be depended upon to stop the crane. To remedy such a condition, we have installed spring bumpers on the cranes and at both ends of the runways in addition to a mechanically operated brake of the correct size for one of the bridge motors. A good crane operator seldom uses the foot brake during normal operation, but in an emergency such as listed above the brake and the bumpers eliminate the possibility of damaging the crane.

Since it is convenient to install a mechanically operated brake on a bridge motor where it can be used when necessary and where it does not interfere with ordinary operation, I see no necessity for considering magnetic brakes on this type of crane.

(3) On ore bridges, coal bridges and similar types of cranes using "Man trolleys," the bridge brake cannot be mechanically operated since the operator travels with the trolley. A series brake with a de-rated coil to take care of the difference between the starting and running currents would probably be satisfactory. These brakes would set, of course, on the "off point" of the controller.

This type of brake was used on our ore bridges several years ago, but we had trouble with the brake coils which were built in our shops. At present we use shunt wound brakes on these motors, arranged to set at the "off point" of the controller.

(4) Motor mounted brakes can be used advantageously where floor space is restricted or where clearances prohibit the use of floor mounted brakes. A good example is the door operating mechanisms of ore transfer cars. This mechanism was designed to be mechanically self-locking, but due to wear in the various joints occasionally dropped a load of ore. A type of motor mounted brake was installed and this trouble was eliminated.

Another advantage of the motor mounted brake is that it eliminates the necessity for a separate rigid foundation for the brake. The motor mounted brake must be removed, however, when motor clearances are small and the motor as a unit must be removed. If an adapter is required for mounting, the cost of a motor mounted brake is higher than the floor mounted brake.

(5) The benefits of systematic inspection and accurate adjustment of magnetic brakes cannot be over emphasized. A poorly adjusted brake is not only a liability as far as braking effort is concerned, but is a hazard to the workmen whose lives may be endangered by the failure of the brake to function correctly in an emergency.

To properly combat the evils of carelessness and ignorance, the following suggestions are made:

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(1) Blue prints of the various makes of brakes should be made with the necessary notes giving the proper procedure for correctly adjusting the brake. All these prints should be framed and mounted in each motor inspector and millwright office.

(2) These maintenance men should be sent, in small groups, to the Electrical Department laboratory several times a year, and be required to correctly adjust the different makes of brakes which are in use in their department. These measures should ultimately eliminate the habitually careless workman who refuses to take advantage of the opportunities which are offered him.

(6) While cast iron has the advantage of a self-lubricating, non-grabbing, and non-freezing surface when used in brake wheels, it should not be used in large high speed wheels, due to the possibility of breakage.

Cast steel and rolled wheels are preferred from a safety standpoint. We now have several trial installations of hardened steel wheels to determine how their life compares with that of standard wheels.

By W. L. WHITE*

(1) Crane trolleys should be built and equipped for plugging with a shunt brake of sufficient size to keep loads from swinging trolley. A series brake coil wound to stand starting current peaks will let brake close after current has died down, due to machine being under way.

(2) On a heavy crane bridge whether one or two-motor drive, if equipped with anti-friction bearings, air brakes would undoubtedly be superior to any other type of brake. A magnetic brake on a crane bridge must be adjusted to give crane a reasonable drift, otherwise it interferes with the work. Emergencies arise when it is necessary for an operator to stop his crane more quickly than his magnetic brake will do it, and in this case his only alternative is to plug the motor which action opens his brake and makes him wholly dependent on plugging. If equipped with air, a heavier application than normally used would give emergency stopping or he can plug the motor along with application of the brake. Have often thought that a disc type brake on bridge armature shaft, manually operated for cranes up to 10 tons or 15 tons should make a very satisfactory brake.

(3) Cranes on which the operator travels with the trolley are usually what we would call a heavy crane such as ingot strippers, soaking pit cranes, and charging machines. On a crane of this type, an operator is helpless in case of power failure. Air brakes could be provided through use of solenoid valves for emergency stops, but for normal operations, plug the motors.

(4) An extension on bottom half of mill type motors with pads to be machined for different types of shoe brakes might be an advantage in the way of maintaining brake alignment over a period of years.

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By A. J. ACKER*

Since our company has developed and has in operation, foot brakes for the bridges of large cranes where the operator's cab travels with the trolley, I believe that would most interest you gentlemen. A length of tiller rope is stretched along the bridge, very much after the fashion of a wire cross conductor. One end is solidly anchored to an eye-bolt and the other end is attached to a bell-crank arm as indicated. In the cab the appearance of the foot brake is exactly the same as on the usual type of crane. It connects, however, as shown, with one of a nest of three little sheaves over which the tiller rope passes. It will be noted that stepping on the foot lever pulls the one sheave up and away from the others, thus shortening the rope and applying the brake. This brake, then, gives real "expression" to the control of the bridge just the same as is obtained with the usual type of foot brake.

As has been pointed out in the papers, a solenoid or magnet operated brake on a bridge motor is not exactly satisfactory. If the magnet brake is set for a light torque then it is of practically no use in an emergency, and if it is set for a heavy torque, approximating the full load torque of the motor, it stops the bridge too promptly every time, taking away all possibility of drifting, but I need not elaborate on this—you gentlemen understand and appreciate the beauty of having a real foot brake on a crane bridge, even though the cage travels with the trolley.

The number of cranes with the cab on the trolley is not great, as compared to the total and therefore, the demand for this brake cannot be said to be great, but we have it in operation on a considerable number of large cranes. One I have in mind right here in the Pittsburgh district has a trolley travel of 129 feet. It is a heavy Gantry Crane where the trolley and operator's cab pass through the legs at each end. This is a heavy crane and has a 135 HP. bridge motor and this type of foot brake has given entire satisfaction on it.

On cranes using two bridge motors, there seems to be no question about the desirability of having a brake on each motor. It is comparatively simple to do this and not at all difficult to equalize the two brakes.

With regard to the other points listed on today's program, our practice is as follows:

On roller bearing trolleys we make provision for a magnet operated brake on the traverse motor, that is, we extend the motor base and the motor shaft so that a brake can be installed at any time it proves to be desirable. It might be said that a brake is not necessary on a roller bearing trolley but only a device which will put back into the trolley when the power is turned off, approximately as much friction as is present in a sleeve bearing trolley. Therefore,

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a brake which will give approximately one-half full load motor torque is sufficient. In general, our practice has been to use a series wound brake. It of course applies every time the power is turned off and if it is of light torque, giving just enough friction or drag to make the trolley drift about like a sleeve bearing trolley would, then it gives the operator the same control over the load that he is used to on a sleeve bearing trolley.

On the question of motor mounted vs. floor mounted brakes, we have not taken sides. In some instances mounting the brake on the motor makes it possible to use a narrower gauge trolley than would be required for floor mounting, and this of course, means less weight and a more economical design.

The necessity for regular inspection of brakes does not need emphasis here. The inspector should, of course, understand the brake. Every now and again we run into a man who tries to adjust for lining wear by taking up on the spring tension. It is important that clear instructions be furnished with every brake, or better yet, that the brake be so simple of adjustment that no study is required to understand it.

Under the heading of the "Best Material for Brake Wheels," it has already been stated that the manufacturers of brake linings recommend cast iron rather than steel. The material worn from a cast iron wheel falls off as so much dust, whereas, the material on the face of a steel wheel drags and smears, becoming very rough and causing very rapid wear of the brake lining. Of course, the next step is the use of a "Tool Steel Wheel," a steel wheel with hardened face. The life of such wheels is very much greater. This whole question of brake wheel wear hinges on the unit pressure which the brake design gives between the lining and the wheel. It is, of course, important to keep this unit pressure down to a conservative figure.

The Theory and Practice in Connection With The Application of Brakes To Motors, For Cranes, Screwdowns, Mill Tables, etc.

Discussion

Discussed By

H. W. Ball, Engineer, Morgan Engineering Company, Alliance, Ohio.

James Farrington, Electrical Superintendent, Wheeling Steel Corporation, Steubenville, Ohio.

F. O. Schnure, Electrical Superintendent, Bethlehem Steel Company, Sparrows Point, Md.

J. P. Biggert, Sales Engineer, Tool Steel Gear & Pinion Company, Pittsburgh, Pa.

George A. Kaufman, Electrical Engineer, Jones & Laughlin Steel Corporation, Pittsburgh, Pa.

Joseph A. Burg, Mechanical Engineer, Electrical Department, Carnegie Steel Company, Munhall, Pa.

W. B. McMasters, Master Mechanic, Wheeling Mold & Foundry Company, Wheeling, W. Va.

H. E. Hodgson, Mechanical Engineer, Cutler Hammer, Incorporated, Milwaukee, Wis.

T. S. Towle, Sales Engineer, Cutler Hammer, Incorporated, Pittsburgh, Pa.

L. A. Watson, Chief Engineer, Clark Controller Company, Cleveland, Ohio.

W. C. Kennedy, Sales Engineer, Cutler Hammer, Incorporated, Philadelphia, Pa.

J. H. Hall, Electrical Engineer, Electric Controller & Manufacturing Company, Cleveland, O.

R. M. Bayle, Steel Mill Engineer, Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa.

C. M. Myers, Electrical Engineer, Youngstown Sheet & Tube Company, Youngstown, Ohio.

W. C. Raube, Industrial Engineering Department, General Electric Company, Schenectady, N. Y.

C. N. McDavitt, Engineer, General Electric Company, Schenectady, N. Y.

H. W. Ball: We, as crane builders, have to stand in with the brake manufacturers and therefore we cannot say very much against brakes. I would like to see them keep the number of wearing parts as small as possible, also the overall dimensions of the brake as small as practical.

The majority of cranes we build, the customers require very close horizontal hook approaches to runway rail and we have to write quite often to get information on the brake dimensions to enable us to be sure of clearing the knee braces to obtain the desired hook approaches. The hook approach to runway rail seems to be the main point, and sometimes the brakes will not permit us to obtain these approaches.

We are anxious to learn about the wearing of the brake lining and the wearing parts, because the customer can enlighten us more on this subject than the crane builder.

On crane bridges with anti-friction bearings we had to increase the size of brakes considerably and we usually use only one brake to a bridge motor,

regardless of the capacity where double bridge drive is required. In a few instances our customers request two brakes for double bridge drives.

They are now beginning to ask for brakes to be operated either hydraulically or air, and we are now working along these lines.

On brake linings, I have had some idea of using a brass strip at center of the lining to help to polish the wheel and also thinking of using cast steel wheels about .45 carbon. Of course if hardened wheels are used on cranes I do not believe it will be necessary to provide any means for polishing the wheel. I do not recall any case where hardened wheels were asked for.

James Farrington: Answering the question as to whether to use shunt wound or series wound brakes on crane trolleys when equipped with anti-friction bearings, I rather favor the shunt brake. On our bridges where the operator moves relative to the bridge, we have used the shunt brakes exclusively.

Where two motors are used on crane bridge drive, we made the mistake of applying only one brake. On a 250-ton ladle crane we put into service February or March of this year, we have worn out one complete set of cast steel brake wheels, and have now started on the second set. We are now using one of tool steel, one rolled steel, and one hardened steel, but the last one was just put in two or three weeks ago. We hope we won't have to replace any of these for at least two or three years.

In answer to the third question, we have applied shunt wound brake to be used in case of failure of power. We feel the operator can control his bridge much neater by his controller, and only in case of failure of power is the brake to set to save him from running into the end of building.

We feel that floor mounted brake is preferable due to the fact that it will give you a more satisfactory mounting. If you want to change to another make of motor after brake has been installed, you are at liberty to do so. If you want to change the whole motor in case of break-down, you do not have the extra care and time consumed in changing brake mounting.

As to proper inspection: We inspect our brakes once a week in connection with weekly inspection. They are inspected when the regular man inspects limit switches and cables every week and he makes a report in his weekly letter to the office. We are thus assured the brakes are inspected at least once a week.

As to the merits or demerits of cast iron or steel brake wheels. We believe tool steel, though it may have less coefficient of friction, will give us better service than cast steel; and I beg to differ with a crane representative this afternoon, in that we do not believe cast iron should be used at all.

F. O. Schnure: Our Association congratulates itself so far on the meeting on brakes. The manufacturers have given their ideas on the subject and it is up to us to continue the discussion from the users' standpoint and give them our ideas. I am sure we all gain a lot by exchanging ideas, at least I know when any particular practice or condition is under fire in our department a general discussion in our foremen's meeting will generally bring out the weaknesses.

Foreman are inherently knockers, and perhaps that is our weakness too. I want manufacturers to bear with us if we appear to be "knockers" this afternoon.

One of the points we operators sometimes overlook when ordering brakes is that when we order, say, a 60 HP motor we also order a 60 HP brake. We have overlooked the fact that we generally motorize one or two sizes larger than actually required for normal operation. Then, when we get this brake in operation we find it will not stay released when running at light loads. We should specify winding more in line with the average load we expect on the motor.

We have not had very much experience with roller bearings on bridges and trolleys. We have recently placed in service three or four general purpose cranes operating with roller bearings, and so far see no reason for brakes.

We have some ladle cranes in the course of erection with roller bearings on bridge and trolley and while we have not put brakes on, we have made provision for them. I do not see how we can use a brake on a ladle crane unless we use a shunt brake or a series brake with a special series winding, either one to have low torque characteristics.

Referring to the questionnaire I do not believe it is necessary to use electric brakes on bridge motors, except for the purpose of preventing an outside crane being blown away by high winds in case of power failure or on any crane where the operator travels with the trolley. On our outside cranes we use shunt brakes to stop the crane in case of power failure during high winds, and on inside cranes we depend on foot brakes altho I do not think they are used very much on general operation. The average operator will not put himself to the extra trouble to operate the foot brake and with good control plugging usually allows faster operation of the crane. In some operations, for instance, in a tin plate plant it is necessary to maintain very smooth operating foot brakes, because loose plate is handled in stacks and in annealing boxes.

In our Steelton Plant I believe Mr. Reed has applied, air operated brakes on crane bridges, which I have not seen, but understand they operate very smoothly. The foot brake lever operates much on the same principle as the foot throttle on your automobile. The outfit consists of a small air compressor mounted on the bridge with the brake, which is air operated. Time will tell whether they are an improvement over present methods, altho I am a little suspicious of the maintainence problem of small air compressors.

Like Mr. Farrington we believe in floor mounted brakes, that is we do not believe in hanging them on the motors. We think it is better to have the brake a separate unit. Brakes, we believe, should have the same inspection as the other electrical equipment on a crane.

Regarding brake wheels, we have operating in the older part of our plant, a very large number of cast iron wheels.

They are built with a heavy rim and a rather small diameter. They add a lot of inertia to the armature which we believe is the worst feature in their application. We have not had any serious

cast iron brake wheel failures and they certainly do wear fine. We have found that cast steel wheels wear rapidly. We use principally, the rolled steel wheel. It is my personal opinion that the rolled steel wheel, in ratio of wear will give four to six times the life of cast steel wheels and that heat treated wheels will have almost an indefinite life. In cases where the brake is worked hard I certainly believe the heat treated wheel is worth the extra cost. Referring to the brake windings we prefer to have the flexible leads, well soldered or brazed to the winding proper, as we have always experienced trouble, no matter what size brake, where a set screw connection was used.

J. P. Biggert: We have made brake wheels of various sizes and for different services according to customers' drawings and in some cases the rim and web thickness, were so thin in proportion to the hub, the design was rather difficult both from the standpoint of making the steel castings as well as heat treating and if the rim and web thickness could be increased, in such cases referred to above, it would be more desirable from the manufacturing standpoint.

We manufacture brake wheels from low carbon steel blanks, which are machined all over for balance, the wearing face specially hardened and toughened to a depth of one-third the rim thickness, allowing two-thirds tough center to back up the hard wearing face, which is ground after hardening and testing from 85 to 90 scleroscope hardness.

From the standpoint of life, we have made numerous tests that have shown from 6 to 7 times greater life than untreated forged steel wheels, without showing appreciable wear.

Geo. Kaufman: We had an installation at Aliquippa that was very hard on brake wheels. When this mill was first started, we had cast iron brake wheels which were used on MD 109 motors. These wheels lasted about two months and had to be refaced. Later the cast iron wheels were replaced with cast steel wheels. These lasted about six months and had to be replaced. In 1927 tool steel brake wheels were installed and up to date the wear has not been appreciable. About every month or six weeks the linings are brushed to remove steel scale that adheres to the lining. The lining appears to wear well and the brake wheels show no appreciable wear.

Joseph A. Burg: Our experience has shown that most of our foremen prefer the series wound brake. The shunt brake is a little more sluggish. Recently we installed four cranes with series brakes on the trolley and the operators think they are fine.

I am interested in what is the best application on a two motor bridge, and would like to hear some comments on whether other operators have found the air operated brakes on both motors superior to the electrical brakes.

Regarding our brake wheels, we have gone to forged wheels, heat treated on all sizes from 21" down. We obtain very good life on the wheel and lining on these. We have some wheels in use over two years and they seem to be as good as the day they were put in. The untreated forged wheels we found scored pretty rapidly and the life of the lining was not very good.

Regarding the rivets used in brake lining, we tried copper, brass and aluminum and finally decided on copper. We never found out why the brass or aluminum did not work so well in our plant. I do not know what the brake builders prefer. Are copper rivets standard?

Aluminum rivets were not a success in our operation. We tried both aluminum and brass and found that they did not do as well as copper rivets. The brass rivets break easier. We have found particles of steel imbedded in the heads of the brass rivets.

H. E. Hodgson: We ran a short test with a lining of rivets until we cut through all the rivet heads, using aluminum, brass and copper. There seemed to be no difference in performance under those conditions and as a consequence use copper rivets as a standard proposition.

Geo. Kaufman: Do crane operators find trouble in correcting swinging loads? Will someone say something on that?

Jos. A. Burg: Our men have had no trouble from this source. It seems to me that this is something that the crane operators have to get use to. We have found that they can control the swing very nicely. They would not want to do without the trolley motor brake now. If it is not working properly we soon get a request to have it fixed.

Geo. Kaufman: It looks like it is a question of learning how to operate to control that, although I imagine that adjustment of that brake would have a lot to do with it

W. B. McMasters: I would like to ask what kind of wheel was used when the test was run on using the different kind of rivets. From experience we have had, there was quite a different result, especially with soft steel wheels. Copper would pick up steel on the head of the rivets and score the wheel badly, whereas brass did not pick up steel or cut the wheel.

H. E. Hodgson: We did not run an extensive test, but ran .45 carbon without hardening, and there was no cutting under those conditions. It has been my observation that when cutting of the wheel occurs, it occurs outside the rivet line rather than inside, and for that reason I have always decided that the rivets themselves had little to do with the wheel cutting. This has been shown by a variety of brakes over a number of years, and in general, wear was greater outside of the rivet line rather than inside.

W. B. McMasters: The trouble may have been due to our working conditions. In our case the wear was occurring in the line of the rivets. We used to use cast iron brake wheel and we had no trouble with copper rivets, but changed to steel and had trouble. I noticed on automobile linings they use brass exclusively and where persons have had to reline their own brakes after coming from the factory and used copper, they found it scored the drum pretty badly. That is what got me to thinking, we were running into our trouble on account of using copper on steel and then we used brass and had no trouble.

H. E. Hodgson: This concern extends the shank of rivets and by careful selection of sizes the rivet can be merely driven into the hole without any

riveting over. That is an advantage we have not used on brakes as yet, because they are generally harder to get. I mean on brakes we have fewer rivets in the shoe relatively for the number of square inches of lining and feel brass is better for riveting over on the back of shoe.

T. S. Towle: I can bear out Mr. Hodgson's statement that one often finds less wear of the wheel under the line of rivets. In a Pittsburgh mill I have seen cast steel brake wheels—not hardened—which showed little or no wear under the line of rivets and with the wheel surface badly scored and worn away outside of the line of rivets. I have seen where wear would come on the lining and not on the rivet.

L. A. Watson: First I want to thank you for the invitation to attend this meeting and hear the reading of the papers on shoe type magnetic brakes. I will attempt to explain some of the features of the split band type brake.

One of the other speakers today gave some data on pressure per square inch between brake wheel and lining as the determining factor of wear on brake wheel and lining. This data indicated that a pressure of 15 lbs, per square inch caused very little wear and that 30 lbs. per square inch caused very excessive wear. In the split band brake we cover approximately 90% of the face of wheel, and thus have nearly twice the braking surface of the shoe type brake. Therefore our pressures are less in that proportion. As a matter of fact, we never exceed 15 lbs. per square inch on any size brake, when developing the rated torque.

Therefore, we should show considerably longer life of wheel and lining, as the pressure is not more than 15 lbs. per square inch, which was stated this afternoon as a very favorable working pressure.

We also claim a lower temperature of lining due to the greater area of lining. It is evident that for a given retarding torque with same size of wheel the total heat will be the same. But in case of the split band brake this heat is distributed over a greater area of lining.

As the temperature and pressure are the determining factors of the wear of wheel and lining, we claim an advantage over the shoe type brake. This we claim as the first requirement of an ideal brake.

In regard to the torque ratings of our brake, we could rate the brake higher, but it has been our experience that as you increase the pressure per square inch, you automatically decrease its life, so that we have adopted a torque for each size of brake comparable to the shoe brake. On that basis we claim longer life per set of lining.

On the subject of brake wheels; our brake wheels are rolled steel of .65 to .75 carbon and .8 manganese. They are about as hard as we can machine; in fact, we sometimes get them so hard we cannot machine them. We have been very successful with hardened and toughened steel wheels in extremely hard service. My opinion is that the hardened and toughened wheel is so much better for severe service applications such as screwdowns, it would pay the mill operator to buy the more expensive wheel.

In going over some of the questions here on whether to use shunt wound or series wound brake on crane trolleys, I do not know that I am in a position to advise you which is the better. We furnish both, and I think that the series winding is in the majority. We do, however, put special windings in trolley brakes. We put in as many turns of as small a wire as we feel we dare, to keep the brake from releasing when the load is light.

When two motors are used on crane bridge equipped with anti-friction bearings, we have several different brake applications in service. One system is with one series brake, set with about one-third torque on one motor and allowed to set each time the controller is brought to the off point. The other motor is equipped with mechanical foot brake, and mills that have installed this system are well pleased with the operation. With this application over size brakes should be used.

Another system is to use two continuous duty shunt brakes, one on each motor, brake set on voltage failure or to be operated from master controller. Controllers must have drift point between off point and first running point. When stopping bridge by plugging, brake will remain open when passing off point.

In the past two years we have equipped several cranes with air-operated brakes. This system we believe to be necessary for good operation on hot metal cranes with anti-friction bearings. We use Westinghouse Traction Brake Company automotive air brake system attached to our mechanical brake.

On cranes where the operator travels with the trolley, we know of no better system than the torque motor operated brake. In this type of brake there seems to be just enough time lag in the setting of the brake to plug the motor to stop without setting the brake. This type of brake will not set fast because the operating spring must drive the torque motor back to the release point. Most applications have been on A.C. cranes. One test application has been made on D.C. cranes.

As to the advantage or disadvantage of motor mounted brakes, I have very little to say on that subject. Not being motor builders, it is rather difficult for us to get necessary dimensions for building the mounting brackets. One of the most serious objections we have heard is the difficulty in keeping the bracket mounting tight.

The inspection of brakes is essential and the main point for the inspection of the brake, is inspection of the air gap. It is not uncommon to be called out on a brake application that is more or less of a failure, to find the air gap on the brake two or three times what it should be. A long air gap is a disadvantage not only when releasing the brake, but when the brake sets. Due to the long travel of the armature, the brake sets with more of a shock, which may cause the load to stop with a jerk.

As to the question of cast iron brake wheels, we do not use cast iron brake wheels. We believe the safety committees of the steel mills are against the use of cast iron, and we have never attempted to build brakes with cast iron wheels.

W. C. Kennedy: In connection with the question of grooving of wheel surfaces we have had occasion to watch this action very closely on magnetic clutches.

A magnetic clutch is a machine which is prac-

tically the reverse of a magnetic brake in that pressure is obtained between two friction surfaces by means of pull of a suitable magnet whereas on a magnetic brake the friction between the surfaces is obtained by means of a spring or other device and the magnet is used to release the same. Otherwise, the two structures are very similar.

The friction surfaces employed on a clutch are practically the same as a magnetic brake, consisting of one surface made of mild steel, against a friction lining of the same structure as employed in the brake.

We had occasion in one instance to watch the operation of a particular installation for a period of nearly two years. The steel surface was actually grooved to a considerable depth while the friction lining showed hardly any signs of wear. The final result after this period of time was that the steel surface wore out completely during the life of one friction lining. The grooves across the face of the steel showed no relation to the location of the copper rivets, which was of particular interest at the time.

This action between friction surfaces sometimes shows surprising results, which we feel as yet, are not clearly understood.

J. H. Hall: We provide transverse slots in the molded linings to collect dirt and steel dust from the face of the wheel. These also serve to allow the linings to fit accurately on the finished inside surface of the brake shoes and thereby obtain a full bearing surface between the linings and the brake wheel. Several of the operating men here present have mentioned that they secure the best results with cast iron wheels. I believe this to be true, because during the application of the brake a higher temperature occurs at certain spots between the lining and the wheel, resulting in the breaking away of small heated particles from the wheel face. On soft steel wheels the hot spot drags over and welds to the wheel or imbeds itself in the lining. This produces cutting and grooving of the wheel rather than the wear of the rivets in the lining. The matter of rivet cutting has been well covered in the discussion, and the absence of rivet cutting is in accord with the results we have observed.

With tool steel wheels hardened and heat treated the carrying over of metal particles is eliminated and the wheel face develops a bright surface which adds much to prevent wear on linings.

R. M. Bayle: We find that the semi-steel wheel from an economic standpoint probably meets the average requirements of the trade. We are prepared to furnish heat treated steel wheels, using the well known Nuttall BP process. This treatment, we believe, gives an exceptionally fine product fit for the severest applications.

C. M. Myers: Our experience is somewhat limited in heavy roller bearing cranes, but we have some general purpose, mill cranes with roller bearings. On the average 10-ton mill crane, we use only a foot brake on the bridge, but no brake on the trolley. We do have a full roller bearing crane being built, a 150-ton ladle crane, to have a double bridge drive, that is two-motor drive on bridge; one foot brake, and one shunt brake for emergency in case of power failure will be used. Also have shunt brake on main trolley as a safety feature for pouring metal. We are using that only as a safety measure. Set our shunt brake to hold main trolley in right position so auxiliary trolley will not tend to shift main trolley when pouring. We will have to await results on this ladle crane.

W. C. Kennedy: The discussion this evening has been extremely interesting both from the standpoint of operator and also the manufacturer.

In an application of electric power to a crane; mill table or other similar device, consisting of a motor, controller and electric brake, the brake has really been given less attention than the other two items. The brake, however, has been considerably improved and today is the result of much thought and engineering design, which would not have been given to this development several years ago.

I believe, however, that one of the greatest contributing factors to the success and the application of electric brakes is the improvement in the design of the apparatus with which they are to be used, for example—the use of the electric brake on a crane has become increasingly important as the crane grew in size and operating speed. This has resulted in an improvement of the methods of drives and the general installation of say, the bridge and trolley.

A few years ago the bridge motor of a crane was not mounted in a good and substantial manner and did not have the necessary foundation and mechanical support that it does today. The improvement in this part of the crane structure has resulted in a very much better operation of the brake and the connected drive in general.

This brings up another question in regard to the operation of brakes in general. A brake should be chosen with just as much consideration for the purpose in view as a motor. It is not at all reasonable to assume that the same brake should be used on a certain horsepower motor of a given speed, regardless of the application of the motor. In the past I think this point has been neglected, and it has been assumed that a certain horsepower motor would require a certain size of brake. This point has been brought out this evening in that you will notice practically all of the discussions have been in regard to brakes used on the bridge motions of crane, while very little comment has been made on the application to hoists. Practically, all of the applications to the bridge motions, for example, are questions of inertia and speed, as well as the deceleration of the load. For hoist applications on the other hand an entirely different case is presented in that the inertia of the load is not a predominating factor and also the speeds and weights of the load for any given size of motors are radically different where the same motor is used for bridge service.

In other words, more attention should be paid to the application of the brake than as previously been done.

For the reasons outlined above, I do not believe that the question of motor mounted brakes as compared to floor mounted brakes has been given the proper attention. It is not right to assume that a certain size brake is suitable for a certain size of motor regardless of its application, therefore, I believe that brakes should be considered as a separate and distinct item and consequently the floor mounted brake is the proper one to use. Otherwise, it may

be impossible to employ the correct brake on a certain installation, due to the fact that proper mounting pads for a certain size have already been provided on the motor. I think that the brake should be considered separately, with due regard for its application and then mount it separately in a substantial manner on a proper sole plate or other foundation.

I differ with the gentleman who said that a brake should be as small as possible. On the other hand, it should be just as large as possible, because the larger the brake and braking surface—the less wear and less maintenance, and proper braking action will be prolonged.

A brake is a decelerator and as such is just as important a factor as the controller, which provides proper acceleration.

One gentleman has brought out the question of the camber in a big girder, which certainly is interesting with the increased use of roller bearing trolley. On light load the trolley might tend to run from the center to the outside whereas with a loaded ladle the design of the crane girder might be such that it would cause the trolley to run from the outside toward the center. This difficulty has been solved in some cases by putting in a series wound brake on the trolley motor providing light torque, which closed quickly, and produced sufficient holding power to prevent drifting. This is a very striking example where it is necessary to consider the application and not simply put in a brake suitable for certain horsepower of motor.

In general the question of the application of shunt wound brakes, in conditions where it is desirable to get quick release, likewise, quick setting—it is important to consider the design of the coil and in many cases using external resistance or other means so that the coil can be designed for a much lower voltage than line voltage, thereby reducing the inductive counter voltage of the coil to considerably less than the line voltage. The operation can be very much improved in this way, as we have seen in the operation of some magnetic clutches where they are required to perform a certain duty cycle in a fraction of a second.

The writer believes it would be interesting to continue this discussion at some other time and consider the construction and application of alternating current brakes.

Geo. Kaufman: Mr. Bayle, you mentioned something in connection with A.C. motor applications; that A.C. rectifiers and D.C. brakes are sometimes used.

R. M. Bayle: We have made some applications. So far it looks good. The economies are not so good as it costs more than the straight A.C. brake. However, the D.C. brake using rectified A.C. doesn't require as much power. We do not have sufficient operating experience behind it yet to recommend it for universal use.

J. H. Hall: Answering the question as to the prevailing need for magnetic brakes on crane trolleys, the use of brakes has really assumed importance since the advent of anti-friction bearings on trolleys. With sleeve bearings a brake is seldom used on the trolley. The use of a brake at any time is either to slow down a load or hold it. When the conditions are such that a trolley will not stand still a brake must be added to hold it. The bridge girders of the crane are designed with a camber sufficient to level the bridge with maximum load at the center. With lighter loads the trolley tends to run toward the end of the bridge. Roller bearings have so little friction that the trolley will not remain at any given place, and it has been necessary, particularly on hot metal trolleys, to use a friction device to replace the friction formerly provided by the sleeve bearings.

Joseph A. Burg: Is it customary to bush pins with steel bushing? Some would change to hardened steel bushing.

H. E. Hodgson: We use an alloy steel with a very high wear resistance and do not harden it. The pins are case hardened, ground, and chromium plated. The bushing should provide some cushioning effect. In a life test on a brake in which the stroke was twice normal and the spring compression was 50 per cent above rating, the shoe was allowed to apply fifty times per minute against a continuously running wheel. At the end of six mililon operations the wear of the hardened and the alloy steel bushings were negligible and almost identical; therefore, in the interests of best economy the alloy bushing is used.

Joseph A. Burg: What has been the experience with different rivet materials?

H. E. Hodgson: We tested aluminum, copper, and brass rivets against the same wheel by driving them in rows with the heads flush with the lining. The brake was then operated until all heads were worn away. No cutting was found on the wheel surface due to any of the rivets. It has been my observation that when wheel cutting occurs it is outside the rivet line rather than in, and I have concluded that the rivets do not cut wheels. These observations have extended over a number of years and with a variety of different brakes and wheels.

One type of rivet which we are using on magnetic clutches is brass. The shank is grooved and bulged somewhat. By careful sizing of the holes it is possible to drive the rivets into the holes and hold the lining permanently without rivetting. We have used these rivets on brakes because more metal is required to grip the rivet shank than can easily be put into the shoes without interferences and because the locations of rivets in shoes are limited by the ribs of the shoes. For brakes, therefore, we use straight copper rivets and peen them over.

W. C. Raube: The General Electric Company is bringing out a line of A.C. brakes, in which the conventional solenoid is replaced by what is termed an "operator". The operator consists of a cylinder, containing a piston, and which in turn contains a built-in impeller driven by a small 3-phase squirrel cage motor through a splined shaft. The motor is mounted in a vertical position on top of the cylinder casing. The piston is connected to the brake mechanism through a yoke rod. A suitable oil is provided in the cylinder, which serves as the pumping medium and also keeps the moving parts well lubricated.

The brakes are spring set and operator released—that is, the operator motor, when energized, forces

oil under the piston, causing the piston to rise, which, through the yoke rod, compresses the brake spring and releases the brake. When it is desired to set the brake, the operator motor is deenergized and the compressed brake spring forces the piston to its lower position. The "down" movement of the piston when brake is setting is slightly retarded by the stored inertia in the operator motor and impeller, thereby giving a smooth application of the brake.

The impeller is furnished with straight blades and, therefore, the impeller pumps oil equally well in either direction of rotation. Power can be applied to the operator motor at the same instant that power is applied to the main motor on which the brake is used.

In my paper delivered this afternoon I made reference to an "operator" in connection with a tapered torque, foot operated electrical bridge brake. The operator there referred to is the same type of operator I have just described, except that in the previous reference the operator motor was a universal type motor, thus making it suitable for D.C. work.

J. H. Hall: We are furnishing as a standard, hardened and ground steel bushings and pins on EC&M Type WB brakes. The bushings are made of Nitroloy steel and are superior to anything we have tried. They show practically no wear on continuous operation.

C. N. McDavitt: In our line of band brakes the pins are of manganese steel sherardized; of large diameter; large bearing surface and are not hardened.

In our line of shoe brakes the pins are of cold drawn steel, case hardened. The hinge pins for the yokes are sherardized but are not hardened. We do not use any bushings in either line of brakes.

L. A. Watson: We use a high carbon steel pin (Steel spec. SAE 1045). The bearings are of extra size and length and are bushed. The Nitroloy bushings which we use are especially hardened alloy steel which will not rust.

W. C. Raube: On our shoe type of brake we use pins on smaller size brake and when brake is outside we use a stainless steel pin. Most pins of large diameter, and also in new line of band brakes we furnish a small cap so that the pins can be examined periodically.

L. A. Watson: I am not sure that brass is the best material to use in riveting linings to bands or shoes, but we have had better results with brass than copper.

In replacing lining we have observed that the steel will wear faster under the lining where there are no rivets indicating that under the rivets the wheel would wear longer. We are now manufacturing one size of brake where the rivets are staggered over the entire width of the brake wheel.

It is my opinion that the excessive wear is often times caused from grit and steel chips embedded in the lining and on examining the lining we found less foreign material in the line with the row of rivets. So we believe that the counterbore for the rivet has acted as a trap for catching this grit. The staggering of the rivets may work out to a better advantage with the wheel as the entire width of wheel will be subject to the same conditions.

W. B. McMasters: Die grade "Cromol" steel heat treated to 45 to 50 scleroscope is giving very good results in brake wheels and track wheels.

The hardness of 45 to 50 scleroscope in this steel is just about as hard as can be machined, and of course is the same hardness all through.

This steel is giving wonderful results in drop forge dies and other applications where there is hard wear and cost is reasonable.

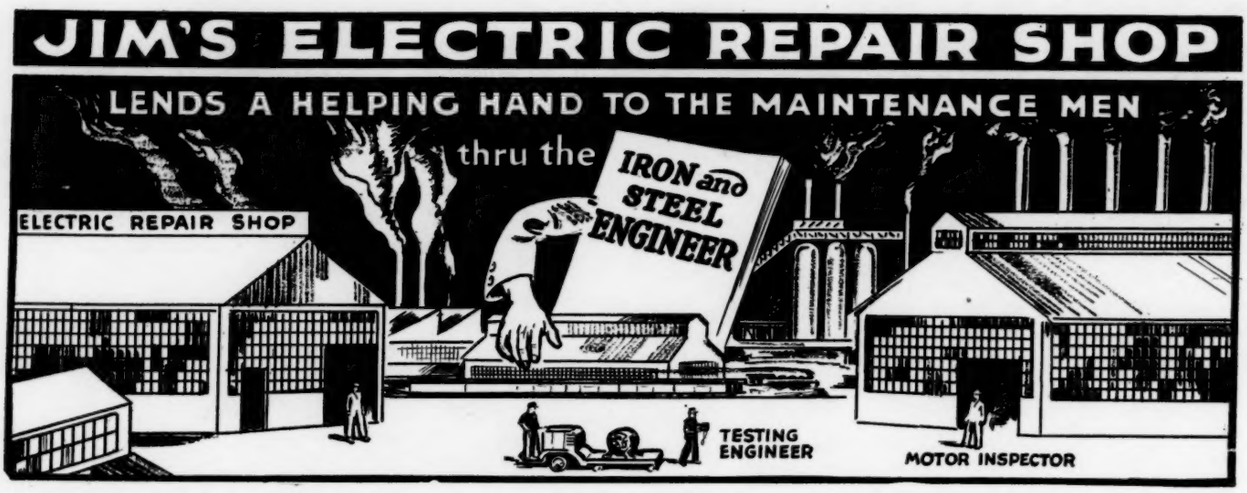
Largest Mercury-Vapor Turbine Will Be Erected In Schenectady

A 20,000-kilowatt mercury-vapor turbine generator is to be installed in Schenectady in a new power plant that will be outstanding in several respects, it has been announced by Burton L. Delack, manager of the Schenectady Works of the General Electric Company. It will be the first outdoor steam plant; and it will be the first power plant ever to have co-ordinated industrial requirements and utility sources of power. The mercury-vapor turbine will be twice as large as the Schenectady-built unit in service at the South Meadow station of the Hartford Electric Light Company; and, because of increased pressure and temperature of operation, it will be even more efficient than the Hartford installation, which itself is so much more efficient than regular steam generating stations.

The new generating station, which will be leased and operated by the New York Power and Light Corporation, will supply electricity for the power company's transmission system and steam for use in the G. E. factory. The four million dollar development will be located on General Electric Company property, on land recently made available for development when a city road was moved from within the property to the Mohawk River bank.

One of the present steam generating plants of the Company, near the front end of the plant, will no longer be used—its twenty old boilers will be dismantled. Steam for the operation of the non-condensing steam turbine in the building will be supplied from the new power station, being conveyed there in pipes in a re-enforced concrete tunnel more than three-fifths of a mile long. The other

(Continued on Page 90)



Many motor inspectors are obliged to go about their task of inspecting the electrical equipment under their supervision lacking the essential knowledge as to what part of the equipment requires the greatest percentage of their time. Their efforts when properly directed will produce results far beyond the expectation of the management toward diminishing maintenance costs and production delays.

Proper direction cannot be effective until a thorough analysis has been made to determine what parts of the equipment causes the most trouble. The operating superintendent can approach this problem by analyzing his past equipment failures and dividing them into six general groups.

We are submitting the following general analysis obtained from a number of Steel Plants in order to develop an educational program which will deal with the types and parts of the electrical equipment requiring the most attention and inspection. It should be equally important, in assisting the maintenance men to analyze their operating conditions and institute an inspection and maintenance program which will improve his operating conditions. Your method of attacking this analysis and the percentages obtained may differ radically from those we have set forth, however, we are offering our analyses, with the thought of "Better Inspection."

The first table shows the general classification of equipment failures:

TABLE NO. I

GENERAL EQUIPMENT FAILURES

Name of Equipment Failures	Failures in Percentage Lost Time	Non Lost Time	Total
1. Motors	30%	50%	80.00%
2. Control	5%	10%	15.00%
3. Wiring	3%	—	3.00%
4. Switching	1%	—	1.00%
5. Power Supply	.75%	—	.75%
6. Unclassified	.25%	—	.25%
Totals	40%	60%	100.00%

We believe this table proves that the major losses in production and efficiency of operation are due to motor failures. These can be further analyzed to show what parts of the motor should be given more careful inspection.

Let us proceed with Table No. 2:

TABLE NO. II
MOTOR FAILURES

Name of Part Failure	Failure in Percentage Pertaining To Motors	To General Equipment
Armature	64%	51.2%
Field Coils	10%	8.0%
Brushes (Traceable Direct)	10%	8.0%
Brushholders and Details	5%	4.0%
Shafts	1%	.8%
Armature Bearings (Hot)	5%	4.0%
Armature Bearings (Other Causes)	1%	.8%
Back Gear Bracket Bearings	1%	.8%
Bolts	1%	.8%
Gears and Pinions	2%	1.6%
Totals	100%	80.0%

The accusing finger points toward the armature which in this table is shown as the cause of 64% of all motor failures and 51.2% of all equipment failures. Before passing we want to call your attention to the fact that this table shows that .8% of all equipment failure are caused directly by armature bearings, excluding lubrication failures. Let us see what future developments have in store for conviction or vindication of the bearing.

Let us scan this 51.2% charged against the armature and see what Table No. 3 has in store for us:

TABLE NO. III
ARMATURE FAILURES

Name of Part Failure	Failure in Percentage Pertaining to Armatures	General Equipment
Commutator	55%	28.16%
Roasted Armature (Due to Overload)	15%	7.68%
Loose Bands	5%	2.56%
Grounded Armature	17%	8.70%
Opened Armature	5%	2.56%
Miscellaneous	3%	1.54%
Totals	100%	51.20%

This table shows three major elements which are contributing factors to high maintenance, namely Roasted Armatures due to overload, grounds and commutator failures. The first is a problem for the Test Engineer to investigate, while the latter two must be further scrutinized to determine the real solution of our problem. Therefore we will present two additional tables:

TABLE NO. IV
COMMUTATOR FAILURE

Name of Part Failure	Failure in Percentage Pertaining to Commutators	General Equipment
Grounded Front V-ring (Due to Oil and Dirt)	60%	16.90%
Grounded Front V-ring (Due to Poor Assembly)	15%	4.22%
Flat Commutators	12%	3.38%
Miscellaneous	8%	2.25%
Short Between Bars	5%	1.41%
Totals	100%	28.16%

TABLE NO. V
ARMATURE FAILURES DUE TO GROUNDS

Name of Part Failure	Failure in Percentage Pertaining to Grounded Armature	General Equipment
Excess Oil and Dirt	65%	5.65%
Loose Coils	20%	1.74%
Poorly Insulated Armature	10%	.87%
Miscellaneous	5%	.44%
Totals	100%	8.70%

Summarizing these tables into one general table each failure and its percentage to the whole will show up if its percentage is high.

TABLE NO. VI
SUMMARIZED EQUIPMENT FAILURES

Name of Equipment Failure	Failure in Percentage Pertaining to		
I MOTORS			
A Armatures			
1. Commutators			
a—Grounded Front V-ring (due oil)	16.90%		
b—Grounded Front V-ring (poor Assm.)	4.22%		
c—Flat Commutators	3.48%		
d—Shorts Between Bars	1.41%		
e—Miscellaneous	2.25%		
Totals		28.16%	
2. Roasting (overloads)		7.68%	
3. Loose Bands		2.56%	
4. Grounds			
a—Excess Oil and Dirt	5.65%		
b—Loose Coils	1.74%		
c—Poor Winding	.87%		
d—Miscellaneous	.44%		
Total		8.70%	
5. Opens		2.56%	
6. Miscellaneous		1.54%	
Total			51.2%
B Field Coils			8.0%
C Brushes (Traceable Direct)			8.0%
D Brushholders & Details			4.0%
E Shafts			.8%
F Arm. Bearings (Hot)			4.0%
G Arm. Bearings (Other Causes)			.8%
H Back Gear Bracket Bearings			.8%
I Bolts			.8%
J Gears & Pinions			1.6%
Total			80.00%
II CONTROL			15.00%
III WIRING			3.00%
IV SWITCHING			1.00%
V POWER SUPPLY			.75%
VI UNCLASSIFIED			.25%
TOTAL			100.00%

It is not surprising to find that the commutator heads the list causing 28.16% of the equipment failures. Since this proves that a great deal of care must be taken in the assembly and maintenance of commutators our next discussion will deal with these features.

Further study of the summarized table shows that 16.9% of the commutator failures and 5.65% of the grounded armatures originate from excess oil. These figures combined with the direct charge of 4.8% makes a total of 27.35% of the equipment failures due to bearings and their care. Therefore, at an early date we will deal with this important phase of maintenance. In the mean time we should receive many suggestions from our readers telling how they eliminated the hazards of excess oil.

QUESTIONS

Hard Drawn Versus Dropped Forged Commutator Bars

I would like to find out the difference in operating characteristics, if any between Hard-Drawn cast and drop forged Copper bars. Large Motor Manufacturers describe in their advertisements that they use hard-drawn copper in their Commutators. Some renewal part Jobbers supply Commutators made from cast copper and drop forged copper. Which material wears longer and has the best polish under adverse operating conditions. C. B. S., Ohio.

Replacing Commutator Bars

What is the most economical method of replacing copper bars in Commutators when it is necessary to replace one or more bars due to a short between bars or to broken bar? J. O. B., Michigan.

Remicaing a Commutator

Are there any refinements in remicaing a commutator which Maintenance Men, the Editor or Manufacturers Engineers can offer which will assist in reducing shorts between bars and excess oil seepages between bars causing the mica segment to be oil soaked. J. H. L., Ill.

May we request that the Steel Mill Repairing, Inspecting and Testing Organizations co-operate and assist "Jim's Electrical Repair Shop" in making our department the most useful part of this periodical by:

1—Submitting your problems to this department for fellow workers suggestions for their proper solution.

2—Submitting your solution to the other fellows problems printed in this department.

3—Submitting your economic methods of performing tasks which will be of interest to your fellow members.

LARGEST MERCURY-VAPOR TURBINE WILL BE ERECTED IN SCHENECTADY

(Continued from Page 87)

generating station within the works, which will be connected with the new station by a tunnel a few hundred feet long, will be retained for some years to come.

In addition to the 20,000-kilowatt mercury boiler and turbine, the new outdoor station will include a steam boiler to supply 300,000 pounds of steam per hour, in addition to the by-product output of 330,000 pounds per hour from the mercury condenser, for process, heating and testing steam in the Works. Adjacent to the power station site is the substation, through which the electric energy from the mercury turbine is supplied to the 110,000-volt transmission system of the power company. It is through this substation, too, that the electricity for the factory is supplied by the power company at 13,800 volts, the voltage at which the mercury turbine-generator will operate.

The design and erection of the new plant are being handled by the Construction Engineering Department of the General Electric Company, of which A. R. Smith is in charge, with engineers of Stone and Webster and of the New York Power and Light Corporation in advisory capacity.

The mercury required in the boiler will weigh a quarter of a million pounds—but so heavy is this liquid metal that such a weight occupies a cubical space less than seven feet on a side.

The mercury-vapor process is an invention of W. L. R. Emmet of the General Electric Company. After preliminary experimentation, a trial equipment was built and operated in the Schenectady plant, from 1915 to 1917. In 1923 a combined mercury and steam power unit was built for the Hartford Electric Light Company. This was the first mercury-vapor power installation in the world. The practical operating experience obtained with this first unit led to the construction of a 10,000-kilowatt mercury-vapor turbine, installed in the South Meadow station at Hartford. The records of the Hartford installation show a substantial saving in fuel over usual steam generating equipments.

Briefly stated, mercury offers decided advantages over water in turbine operation because it boils at a much higher temperature. The efficiency of any heat engine may be increased by increasing the temperature range through which it works—it is for this reason that steam turbines have been constructed to operate at higher temperatures, and higher steam pressures. The properties of mercury are such that high temperatures can be obtained without high pressure. For example, mercury vapor at a temperature of 958 degrees Fahrenheit has a pressure of 125 pounds gauge while steam at a temperature of only 569 degrees Fahrenheit has a pressure of 1200 pounds gauge.

Mercury is boiled and vaporized over a fire just as water is boiled and vaporized in a steam boiler. This mercury vapor then drives a mercury turbine, just as steam drives a steam turbine in an ordinary system. At the exhaust end of this turbine the mercury vapor is still hot enough to boil water and make steam at pressures which are in common use. Therefore, instead of circulating cooling water through the mercury condenser as in the ordinary steam condenser, a level of water is held in the condenser just as in a steam boiler. The water boils and makes steam. A large part of the power generated from the mercury turbine is obtained at very high efficiency since the heat in the exhaust is not lost but is used in the steam production.

For the Schenectady installation the mercury boiler drums will be longer than those of the 10,000-kilowatt equipment at Hartford, but the furnace width will be no greater. The design of the unit was based on generating 20,000 kilowatts from the mercury turbine and 240,000 pounds of steam per hour made by condensing mercury (the additional 90,000 pounds of the 330,000 pounds already referred to is obtained from the water walls). The doubling of the Hartford capacity will be obtained by increasing the mercury pressure to 125 pounds gauge and by installing mercury heating surface on the upper portion of the furnace walls. By thus protecting the furnace walls, the heat liberated in the furnace can be greatly increased over that at Hartford, where the furnace walls are air-cooled. Moreover, the combustion air can be preheated to a higher temperature than was considered desirable with the Hartford furnace. Therefore, a low flue-gas temperature to the stack can be obtained without the use of a water economizer. This will permit a greater application of regenerative feed heating with the steam cycle in case condensing steam turbines are used for generating power.

This 20,000-kilowatt apparatus can be installed in a space no greater than that needed for the 10,000-kilowatt equipment at Hartford.

When pulverized fuel is burned, the lower portion of the furnace walls should for the present be protected by water heating surface. The fuel economy will be comewhat impaired with this arrangement of furnace as additional fuel is needed to generate steam directly in the furnace walls. Further experience is needed before the entire furnace walls can be protected with mercury heating surface.

With a load of 20,000 kilowatts on the mercury turbine and with the power developed from the 240,000 pounds of steam generated at 400 pounds pressure and 350°F. by condensing the mercury vapor, the expected fuel rate will be 8800 B.t.u. per net kilowatt-hour. With water cooling in the lower part of the furnace, the fuel rate will be from 9100 to 9500 B.t.u. per new kilowatt-hour, depending on the amount of steam generated directly in the water-cooled furnace walls.

Hartford Performance Data

In view of the announcement of the large mercury-vapor turbine-generator which will be installed in Schenectady, the following data regarding the performance of the Hartford turbine in 1930 is of interest. The Hartford unit has been in continuous operation since February 4, except for occasional week-ends and during nine days in May when minor changes were made. The ease in starting and operating the apparatus, and the lack of appreciable vibration and noise of the 10,000-kilowatt mercury turbine, is apparent. Low fuel consumption has been obtained consistently over long periods of continuous operation. During the nine months from February to

October, inclusive, the performance of the unit was as follows:

Coal burned	69,187,540 lb.
Water evaporated	572,415,000 lb.
Output mercury turbine	42,297,000 kw-hr.
Equivalent output from steam	56,910,660 kw-hr.
Total station service	1,924,390 kw-hr.
Total net output	97,283,270 kw-hr.
Coal rate on total net output	0.712 lb./kw-hr.
Hours in service	5,030
Use factor	76.6
Capacity factor	64.6

The availability factor of the equipment was as follows:

	Feb. 4 to Oct. 31	July 13 to Oct. 31
Mercury apparatus only	96.6	99.5
Mercury plant complete	83.5	88.6

The kilowatt-hours generated would have been much greater except for excess water power available during the early months of the year and also during nights and week-ends. The tabulation given below shows average daily operation records of the equipment under commercial conditions at full load:

Load on mercury turbine	10,000 kw.
Speed of mercury turbine	720 r.p.m.
Steam produced per hour	129,000 lb.
Feed water temperature to economizer	214° F.
Feed water temperature to mercury condenser	275° F.
Steam pressure	280 lb. ga.
Steam temperature	735° F.
Mercury vapor pressure at turbine	70 lb. ga.
Mercury vapor temperature at turbine	884° F.
Mercury condenser vacuum	28.25 in. Hg.
Mercury condenser vacuum temperature	447° F.
Temperature of air at burner box	460° F.
Temperature of gas to stack	335° F.
Coal burned per hour	15,800 lb.
Heat in coal as fired	14,300 Btu/lb.
CO_2 in flue gases	14.0%

The electric auxiliary power, amounting to some 415 kilowatts, includes water-feed pump, induced and forced-draft fan, coal-pulverizing units, generator excitation, etc. Commercial operation over a period of several days with a steady load of 9500 kilowatts on the mercury-turbine generator gave a net economy of 9800 B.t.u. per kilowatt-hour, and over a period of nine months 10,180 B.t.u. per kilowatt-hour. An improvement in economy can be made when steam is generated at the designed pressure of 400 pounds gauge and the feed water heated to a greater extent by extracted steam.

The ease in operating this apparatus was fully demonstrated in 1930, and experience during the year indicated that maintenance costs will be less than with standard steam plants.

Items of Interest

PERSONNEL CHANGES

F. C. Swartz has been appointed general manager of the Steubenville works of the Wheeling Steel Corporation succeeding E. N. McKelvey whose resignation became effective on February 1st. Mr. McKelvey has been in charge of the Steubenville operations since December 1924 and during his management the Steubenville works has undergone an extensive expansion program.

Mr. Swartz has been assistant to Mr. McKelvey for the past nine months. He took over the management of the operations on February 1. The Steubenville works stands today as one of the largest and most important links in the Wheeling Steel group. Mr. Swartz came to Steubenville, Ohio from Cleveland where he was associated with the American Steel and Wire Company for many years. He is a practical, experienced steel man having been identified with the steel industry for the past twenty-five years. The membership of the Association of Iron and Steel Electrical Engineers extend their congratulations and best wishes to Mr. Swartz in his new connection.

Walter C. Griffith, formerly assistant superintendent has been appointed general superintendent of the McDonald bar mills of the Carnegie Steel Company at McDonald, Ohio, succeeding C. W. Meyers, whose resignation as general superintendent became effective February 1.

Raymond K. Albright has been elected president of the Buffalo Bolt Company of North Tonawanda, N. Y., succeeding the late Ralph Plumb.

J. R. Tanner, formerly vice president and general manager of the Pittsburgh Valve Foundry & Construction Co., Pittsburgh, has been elected president of that company, succeeding C. A. Anderson, Jr., who has severed his connection with the company.

R. L. Wilson, assistant to the president of the Westinghouse Electric and Manufacturing Company, has resigned from that position, after 37 years of active service. Mr. Wilson's resignation became effective December 31, but he will be retained by the Westinghouse Company on call in a consultative capacity.

F. A. Merrick, president of the Westinghouse Electric and Manufacturing Company, announces the appointment of Herbert A. May as assistant to the president. Mr. May will make his headquarters in the Grant Building, Pittsburgh, where the offices of the Westinghouse Central Sales District are located.

On December 31, R. R. Wason was elected president of Manning, Maxwell & Moore, Inc. He succeeded C. A. Moore, who had been the president of this company since April, 1927. Mr. Moore retains his contact with the company as chairman of the board of directors.

Two new appointments, just announced by the Westinghouse Electric and Manufacturing Company,

affect stock control operations in the East Pittsburgh works of the company. A. S. Duncan is appointed manager of shipping stocks and H. C. McElhone is assigned to duties in connection with the analytical reviews of office operations and order procedure.

Arthur G. McKee & Co., engineers and contractors, with headquarters at Cleveland, have established an Eastern office at 120 Broadway, New York City. The new office will be under the direction of Walter G. Mortland of New York. For many years Mr. Mortland has been active in the iron, steel and oil industries.

January 1 Malcolm Carrington assumed his new duties as assistant to the vice president of the Westinghouse Electric and Manufacturing Company. formerly Mr. Carrington was director of finished stocks. In his new capacity, he will have general supervision of shipping stocks, order procedure and office operations.

Clifford F. Messinger, formerly vice president and general manager of the Chain Belt Company, of Milwaukee, was elected president at the December meeting of the board of directors, succeeding his brother, C. R. Messinger, who resigned to become president of the Oliver Farm Equipment Company, January 1.

W. J. Crowe, rolling mill engineer, and L. A. Deesz, fuel and gas engineer, have joined the staff of Freyn Engineering Company and sail January 22 on the steamship Bremen, enroute to the U. S. S. R. to join the Freyn organization of engineers which is constructing a steel works at Kuznetsk, Siberia.

At the January meeting of the board of directors of Harnischfeger Corporation, Walter Harnischfeger, formerly vice president, was elected president and treasurer of the company. He succeeds his father, the late Henry Harnischfeger. The present officers of the company are as follows: President and treasurer, Walter Harnischfeger; vice president, E. Standfuss; secretary, R. Binkowski; assistant treasurer, C. Haugaard; assistant secretary, F. J. Hartmann, Harnischfeger Sales Corporation.

Several appointments of importance, effective January 1, 1931, has been announced by officials of the Westinghouse Electric and Manufacturing Company.

Thomas S. Perkins, formerly manager of Supply Engineering and consulting engineer, is made general manager of Distribution Engineering and will represent S. M. Kintner, assistant vice president, in his work with several departments.

R. W. Owens, manager of Industrial Motor Engineering, has been made general manager of Industrial Engineering and will represent Mr. Kintner in another group of departments.

W. M. McConahey is appointed consulting engineer, with headquarters in the Transformer Engineering Department of the Sharon works. H. V. Putman succeeds Mr. McConahey as manager of the Transformer Engineering Department.

M. S. Hancock is appointed assistant manager of the Industrial Motor Engineering Department and R. E. Marbury is made assistant manager of the Supply Engineering Department.

WITH THE MANUFACTURERS

Surface Combustion Corporation of Toledo, Ohio has recently contracted to install six continuous pack or pair walking beam type, double row furnaces at the Fairfield Works of The Tennessee Coal and Iron Company at Birmingham, Alabama.

The furnaces are of the latest design with an automatic discharge which delivers the pack direct to the mill and eliminates the necessity of having operators at the discharge end. The entire operation being under the control of the Roller. The atmospheric and furnace temperatures are also automatically controlled.

The Allis Chalmers Manufacturing Company of Milwaukee, Wis., are distributing Bulletin No. 1647 on Type "SSU" Centrifugal Pumping Units. The bulletin is well illustrated and contains very interesting tables and charts. Copy may be obtained by writing the above company.

ASSOCIATION NOTES

Two hundred men attended the sessions of the Electric Heating and Welding Conference held on January 21 at the William Penn Hotel under the auspices of the Association of Iron and Steel Electrical Engineers and Pennsylvania State College. The program consisted of nine very interesting papers and the papers and discussions presented at the Conference will appear in an early issue of the Iron and Steel Engineer.

Devices and Appliances for Steel Mill Roll Neck Lubrication will be the subject discussed at the Lubrication Engineering Division's meeting scheduled in Pittsburgh on February 18. The meeting will be held at William Penn Hotel and program will start at 2:00 P. M.

OBITUARIES

It is with regret that we announce the death of C. J. Garrigan on January 23. Mr. Garrigan was electrical engineer of Pittsburgh Rolls Corporation and was an active member of the Association of Iron and Steel Electrical Engineers for the past thirteen years. The membership of the A.I.&S.E.E. extend their sympathies and condolences to the relatives.

C. F. Olin, advertising manager of the New Departure Manufacturing Company of Bristol, Conn., for the past 26 years, died at his home in that city on December 25.

John F. Lewis, former assistant general superintendent of the Edgar Thomson works of the Carnegie Steel Company and a veteran steel worker until his retirement in 1925, died on January 26 at his home in this city. Mr. Lewis was an employee of the Carnegie Steel Company for 50 years. He started in 1875 as a hammer boy. In 1903 he was appointed assistant general superintendent and held that position until he retired in 1925.

A. E. Norris, chief engineer of the Mead Morrison Manufacturing Company, East Boston, died at his home in Brookline, Mass., on January 15. Mr. Norris had been associated with the Mead Morrison Company or its subsidiaries as a mechanical engineer for 35 years.